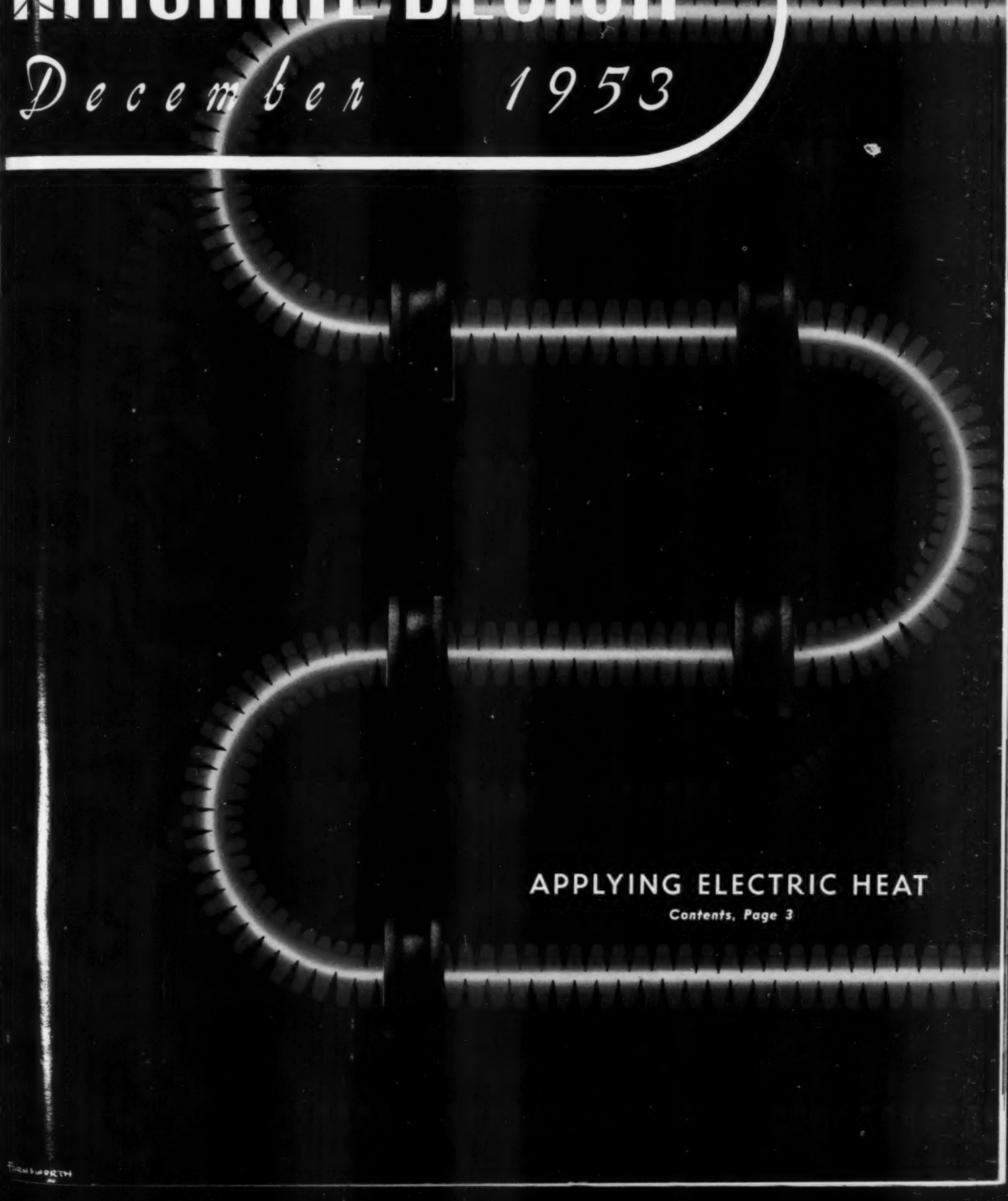
MACHINE DESIGN

December 1953



APPLYING ELECTRIC HEAT

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THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

MACHINE DESIGN

December, 1953

Vol. 25—No. 12

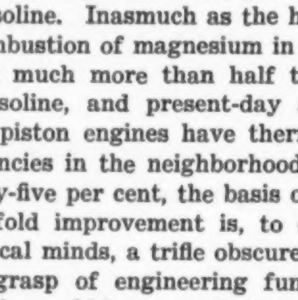
Over the Board

Something for Nothing?

In this day and age no one with any engineering background believes that a perpetual motion machine is practical. But sometimes people can be misled into believing almost equally fantastic claims. You may have read a recent news item reporting the development of an engine using powdered magnesium as a fuel. Now there is no reason why such an engine could not be successful—Rudolph Diesel tried powdered coal in his earliest engines; gas turbines are now being developed to burn powdered coal, and powdered magnesium itself has been used to boost the power of jet engines. But the inventors of the magnesium-burning engine claimed that one pound of magnesium fuel would produce as much power as ten gallons (sixty pounds) of gasoline. Inasmuch as the heat of combustion of magnesium in air is not much more than half that of gasoline, and present-day aircraft piston engines have thermal efficiencies in the neighborhood of twenty-five per cent, the basis of a sixty-fold improvement is, to our skeptical minds, a trifle obscure. A firm grasp of engineering fundamentals would have saved these inventors from such a fluff.

This Month's Cover

For a cold December day we hope you will appreciate the warmth of this month's front cover, which was designed to highlight Associate Editor Leo Spector's article on "Applying Electric Heat" beginning on Page 142. We are sorry that printing schedules would not permit us to depict cooling coils on covers reaching our southern readers.



Two in the Bush

A number of readers have commented on the Editorial "One in the Hand or Two in the Bush?" which appeared in our August issue. Here is one from Vaughan Sanders of Norwalk, California: "After a discussion of your editorial with a number of colleagues, including a supervisor and a group leader, it was apparent we did not all get the same message. The moral, as I saw it, was that unless management faced facts it would be necessary to hire two men for each additional man needed, because the dissatisfaction caused by the unbalance between new and old men would result in loss of the older men. My colleagues did not get this idea from reading the editorial, but agreed that my interpretation might be what you intended."

No, Mr. Sanders, the conclusion you draw was not intended when the editorial was written. However, the moral as you see it is a logical extension of the intended meaning. Let's hope that management never will allow the situation to deteriorate that far.

Annual Index

We continue to receive suggestions from our readers that we publish an annual index of editorial contents, despite the fact that such an index has always been available for the asking and that the index has been printed in our December issue since 1947. This year is no exception, and you will find the index beginning on Page 433 of this issue. You may have an extra copy if you wish, either by writing to us or by using the return postcard facing Page 252.

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MACHINE DESIGN
DECEMBER 1953

Mechanisms—The Vital Link

DESPITE the advent of the well publicized electronics age, mechanical devices—linkages, cams, gears, etc.—are still the backbone of much engineering development. Even the electronic devices themselves, like radar, television and computers, involve mechanical movement of many elements and therefore require mechanisms to complete their function.

Although much has been accomplished in analyzing mechanisms, engineers are still largely dependent on ingenuity and skill in devising a mechanism to perform a specific function. Principles of systematic design and "synthesis" are gradually being developed and understood, but mechanism engineering is still more of an art than a science.

Increased speeds of modern automatic machines, and the high precision necessary in such units as computers, have resulted in a tremendous rebirth of interest in basic mechanism design.

As a contribution to improved mechanism design, we present in this issue the papers delivered at the First Conference on Mechanisms which MACHINE DESIGN co-sponsored at Purdue University this Fall. Interest in the Conference, as measured by attendance and the enthusiasm displayed, was most gratifying and has encouraged us to plan for continuing it on an annual basis. It is our expectation that the published proceedings will grow into a basic reference source on mechanism design that will be indispensable to design engineers.

Regularly in the pages of MACHINE DESIGN, and now through the Mechanisms Conference, we have helped provide design engineers with the latest tools of mechanism design. Response to these efforts has confirmed our conviction that information on improved mechanism design techniques is more urgently needed than ever before. As the vital links between driven elements and their power sources or actuators, mechanisms give a machine its individuality and hold the key to its success. Their design demands mobilization of every technical resource that can be brought to bear.

Colin Carmichael

EDITOR

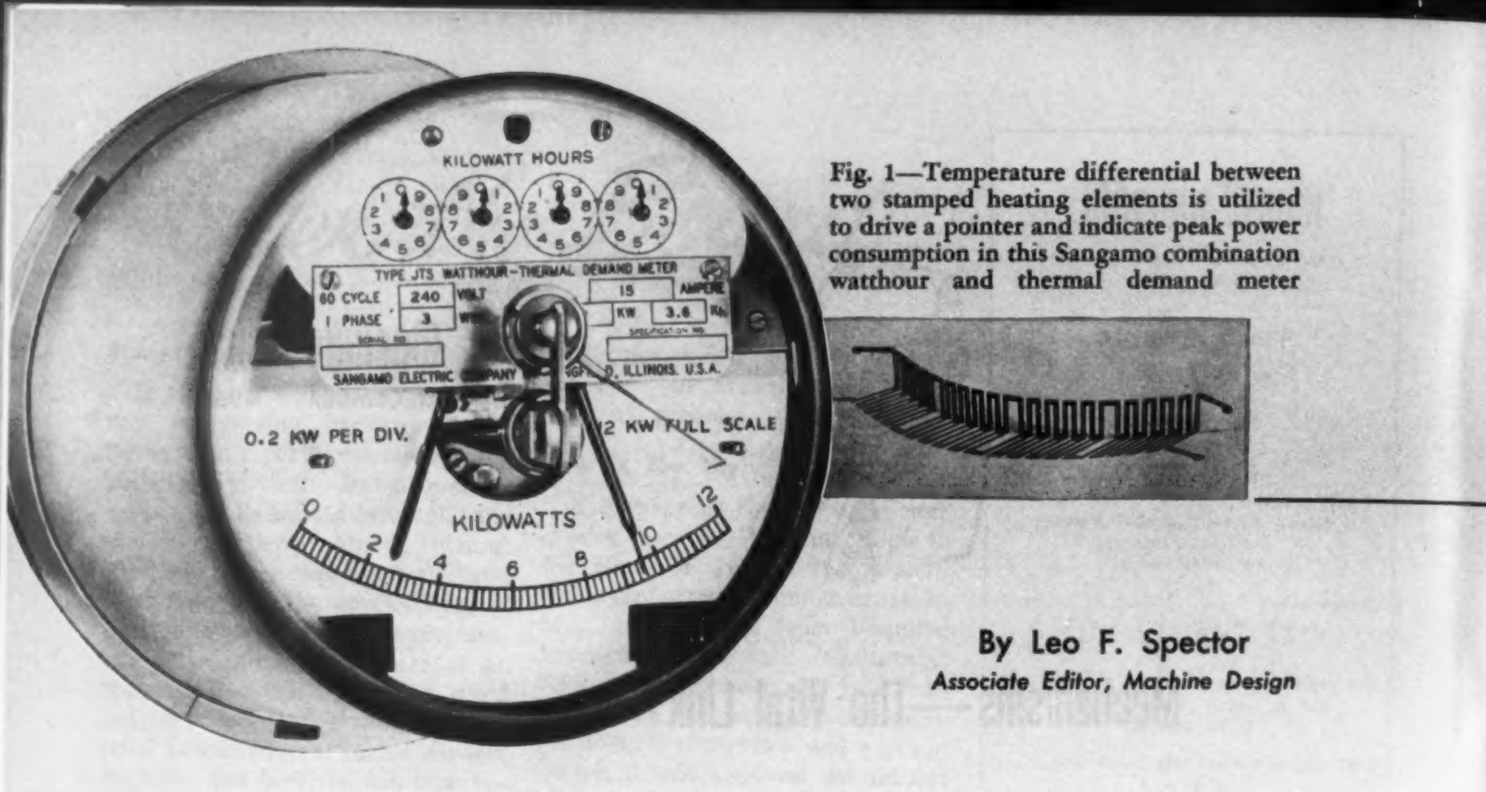


Fig. 1—Temperature differential between two stamped heating elements is utilized to drive a pointer and indicate peak power consumption in this Sangamo combination watthour and thermal demand meter

By Leo F. Spector
Associate Editor, Machine Design

EMINENTLY suited to a wide variety of applications, both industrial and domestic, electric heating offers two distinct advantages; uniform heat distribution and accurate temperature control. Today, this method of energy conversion has been developed to the point where highly efficient applications are being made in machines and equipment ranging from large furnaces to tiny instruments, *Fig.* 1. Modern requirements of greater speed, precision, serviceability, etc., make careful study of electric heating methods a necessity and also serve to highlight their unique advantages in design.

In general, electric heating encompasses a range of methods. These can be enumerated as: (1) Dielectric, (2) induction, (3) resistance and (4) infrared heating. Each possesses particular attributes and today good design research must consider each in proper perspective.

For purposes of this article, one particular area has been selected for discussion—that group of resistance heating devices which can be readily adapted as built-in machine components. In most machine applications, the operating temperatures commonly encountered will fall within the range extending up to 1500 F. Attention will be devoted to those resistance heating units which are specifically suited to this range of operation. Their types and the factors entering into their selection and application as machine components will be covered.

In general, the analysis of any electric heating application can be broken down into three fundamental steps: (1) Calculation of heat requirements, (2) appraisal of basic factors influencing the selection of a heating unit, and (3) evaluation of methods of mounting and installation. Each of these steps in turn involves specific considerations. The value of this simplified design approach lies in a convenient grouping which permits a clear perspective view to be taken of the influence of the different variables encountered. The following discussion treats in detail the contributing factors which must be taken into account.

Determining Heat Requirements: Installed kilowatt heating capacity for a particular application is determined by the larger of two quantities:

1. Initial energy required to bring the material being heated to operating temperature.
2. Energy required to maintain the material being heated at the proper operating temperature.

In each case, calculation of the total heat required should be guided by the characteristics of the specific application and must include allowance for all factors such as heat absorbed by auxiliary equipment, heat losses through exposed surfaces, etc. Dependent on operating conditions, a margin of safety of 10 to 30 per cent should be allowed for contingencies; automatic temperature control can be used to limit the kilowatt-hour consumption to actual requirements. When practical, operating tests offer the best means of determining the necessary heat capacity.

For determining heat requirements, the following fundamental equations are useful in calculating the individual heat quantities associated with different specific applications. Energy required for material heat-up, either gases, solids, or liquids, may be computed from

$$Q_1 = \frac{Wc\,(T_2 - T_1)}{3412\,t} \quad \text{(1)}$$

APPLYING ELECTRIC HEAT

Basic considerations in the analysis, selection and application of built-in units for effective heat control

where Q_1 = energy required in kilowatts; W = weight of material in pounds; c = specific heat of material in Btu per pound; T_1 and T_2 = initial and final temperatures, respectively, of material in degrees F; and t = time required for heat-up in hours.

Heat losses at operating temperature are generally determined by

$$Q_2 = \frac{q_l A}{1000} \qquad (2)$$

where Q_2 = heat loss in kilowatts, q_l = unit heat loss in watts per square foot and A = exposed area in square feet. The quantity q_l is usually found as a chart value. Several useful charts for determining heat losses common to most applications are shown in *Figs.* 2, 3 and 4. A wide variety of these charts for a number of different materials and exposure conditions can be found in handbooks and manufacturers' literature. Care should be exercised in the use of these charts to assure correlation of units with those in Equation 2. For heat losses during heat-up, a reasonably accurate determination of the average loss may be found by taking half of the value at the operating temperature.

Heat absorbed during melting or vaporization is given by

$$Q_3 = \frac{q_p W}{3412\, t} \qquad (3)$$

where Q_3 = energy absorbed by material during a change of state and q_p = heat of fusion or vaporization in Btu per pound. Calculations involving a change of state in the material being heated must include allowance for the alteration in the specific

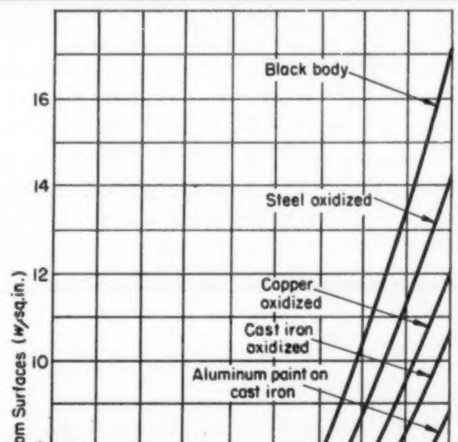


Fig. 2—Chart for determining heat losses from uninsulated solid smooth surfaces at 70 F ambient temperature. Heat losses from bottom surfaces are about 55 per cent of those from top and bottom surfaces

heat of the material which occurs. This variation usually necessitates separate calculations of the heat absorbed by material before and after the point of change.

These three equations are basic for all heat calculations. Summation of the individual heat quantities for specific applications is indicated in the following general expressions for initial and operating heat requirements. Initial heat requirements are represented by Q_i and operating heat requirements by Q_o, both in kilowatts.

SOLIDS: Initial heat requirements include the heat absorbed by the material and the average losses from exposed surfaces. At operating temperature, only the heat losses usually need be supplied. These quantities may be expressed by

$$Q_i = Q_1 + \frac{\Sigma Q_2}{2} \quad \text{(4)}$$

$$Q_o = \Sigma Q_2 \quad \text{(5)}$$

where Σ indicates a summation of the appropriate quantities.

LIQUIDS: Initial heat requirements include the heat absorbed by the liquid and its container as well as the average surface losses from the container and exposed liquid surfaces. Operating requirements include heat-up of added liquid and all surface losses at operating temperature. Thus,

$$Q_i = \Sigma Q_1 + \frac{\Sigma Q_2}{2} \quad \text{(6)}$$

$$Q_o = Q_1 + \Sigma Q_2 \quad \text{(7)}$$

GASES: In most cases of forced convection heating, only the operating capacity need be calculated from Equation 1. A more convenient form of Equation 1 for this purpose is

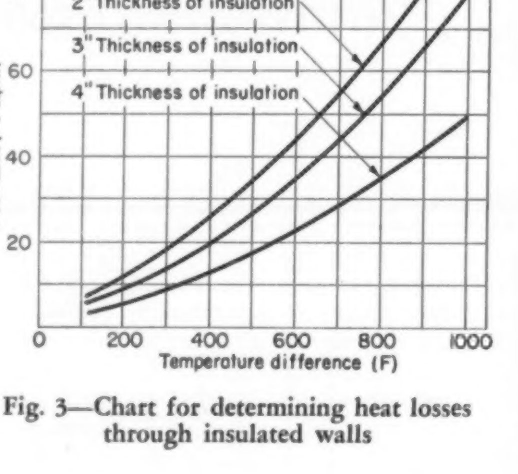


Fig. 3—Chart for determining heat losses through insulated walls

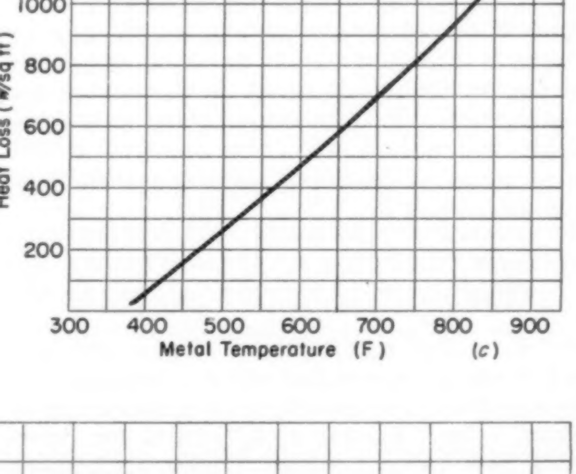
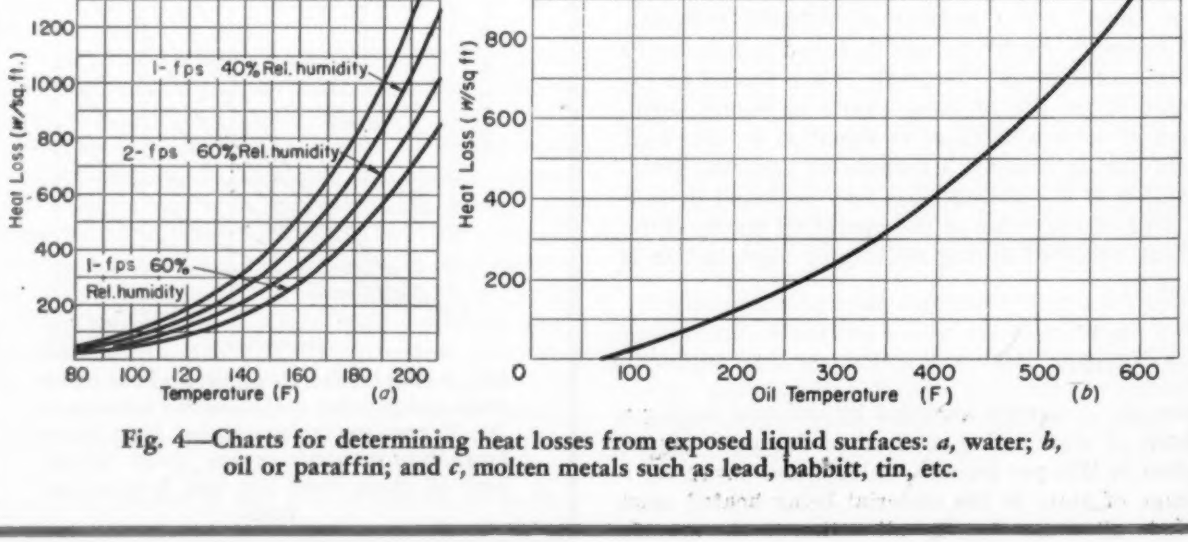



Fig. 4—Charts for determining heat losses from exposed liquid surfaces: *a*, water; *b*, oil or paraffin; and *c*, molten metals such as lead, babbitt, tin, etc.

$$Q_o = \frac{V d\, c_a\,(T_2 - T_1)}{56.9} \qquad (8)$$

where V = gas volume in cubic feet per minute measured at normal atmospheric temperature and pressure, d = gas density at normal conditions in pounds per cubic foot and c_a = specific heat of gas at normal conditions in Btu per pound. Where natural convection, or some form of recirculation is being employed in a confined area, heat losses through the walls may become significant and should be considered. The general relationships indicated in Equations 6 and 7 for liquid heating may be applied under these conditions.

METALS: Initial heat requirements include heat absorbed by metal before and after melting, heat of fusion, heat absorbed by container, and average surface losses from container and exposed metal surface. Operating requirements include melting and heat-up of metal added and surface losses at operating temperature. Thus,

$$Q_i = Q_1' + Q_3 + Q_1'' + Q_1 + \frac{\Sigma Q_2}{2} \qquad (9)$$

$$Q_o = Q_1' + Q_3 + Q_1'' + \Sigma Q_2 \qquad (10)$$

where prime quantities indicate heat absorption of metal before and after melting and Q_1 in this case represents heat absorbed by container. These same general expressions are also applicable to any other heating processes in which a change of state occurs.

These expressions only indicate general relationships for different specific applications. Each case must be analyzed carefully to assure inclusion of all possible factors affecting the heat capacity.

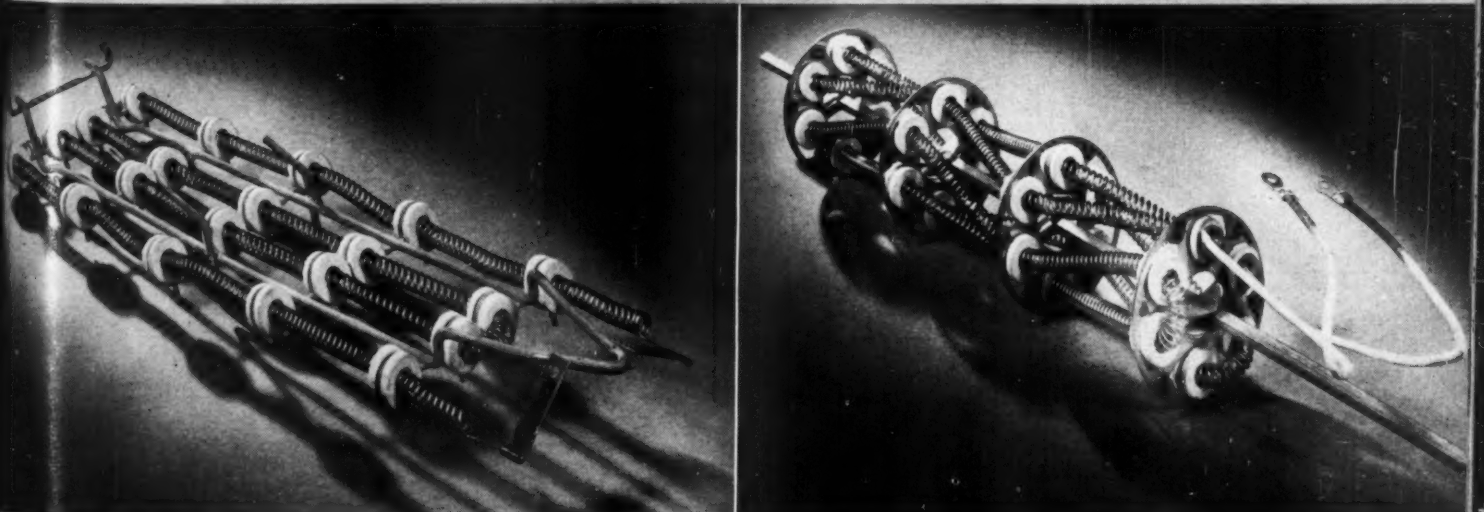
Final specification of heating unit size will be dictated by the larger of the two heat requirements. It must be remembered that initial heating requirements are influenced by the time required for heat-up. Where this time can be increased, heat requirements are decreased and reduced cost and improved performance will usually result.

Open Heating Elements: Wire and ribbon forms of high-resistance alloys have been developed in several grades for use as heating elements at operating temperatures up to 2450 F. Two principal alloys are in use—nickel-chromium or aluminum-iron. The latter, a Swedish development, is comparatively new in this country and is available in three grades for maximum temperatures of 2100, 2370 and 2460 F. The well-established nickel-chromium compositions are also usually separated in three different grades; general practice has established recommended maximum temperatures of 1400, 1700 and 2000 F for these grades primarily on the basis of nickel content. Cost is directly related to the recommended maximum temperature.

Available sizes of ribbon and wire, which is usually supplied in coiled form, will vary with the manufacturer; however, the different alloy grades can be furnished in most of the B&S gages. Resistance and wattage characteristics vary with the grade of alloy; extensive data on properties and characteristics as well as other pertinent specification information are available from the individual manufacturers. The following discussion covers some of the more significant factors which should be taken into account.

Wire elements are generally in coil form and are supported by ceramic insulators, *Fig.* 5. Ribbons are usually wound on a mica form or bent into loops and inserted in slotted ceramic supports. Because of their exposed, uninsulated surface, these elements must be safeguarded against contact with liquids or surfaces. Environmental atmospheric conditions will have a significant effect on element life since susceptibility to corrosive influences in-

Fig. 5—Open type elements for domestic clothes dryers supply 4200 watts at 240 volts ac



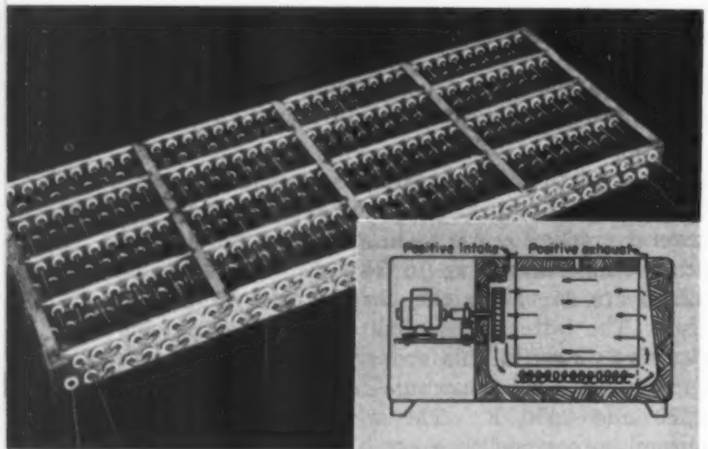
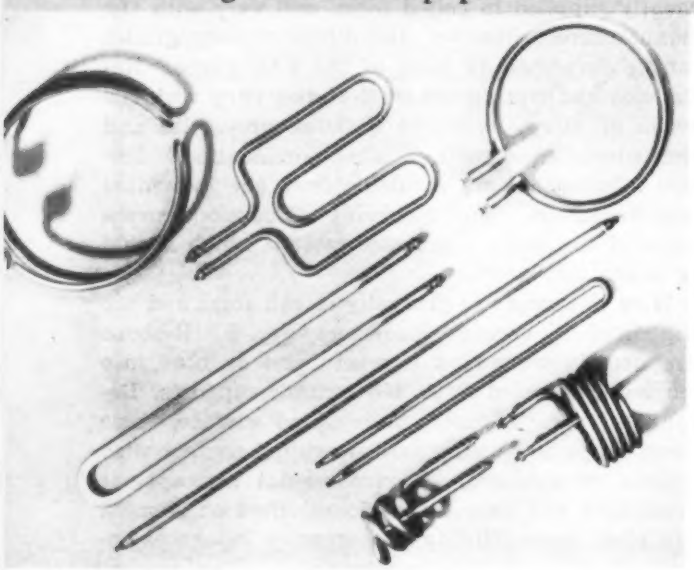


Fig. 6—Above—Con-Wate mechanical convection oven utilizes heater bank of open elements and bimetallic damper to obtain precision control of heat transfer

Fig. 7—Below — Typical enclosed tubular heating units showing some of the shape possibilities



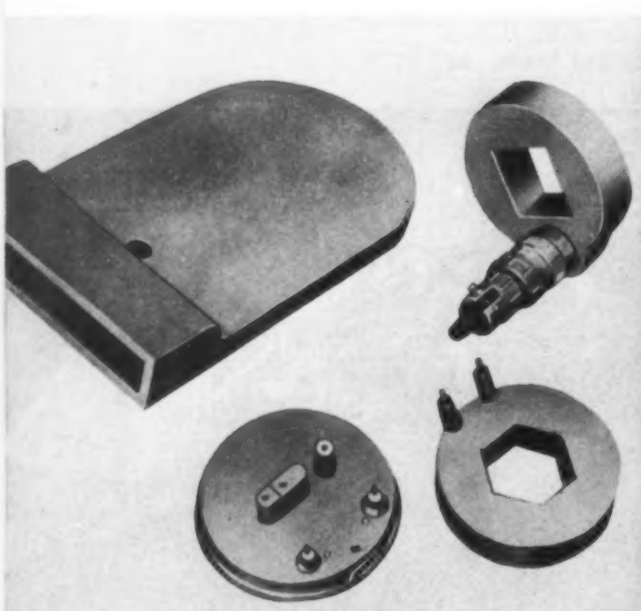
creases with temperature. Low temperatures and intermittent usage provide the best working conditions. In general, life as well as cost is substantially less than for the enclosed elements.

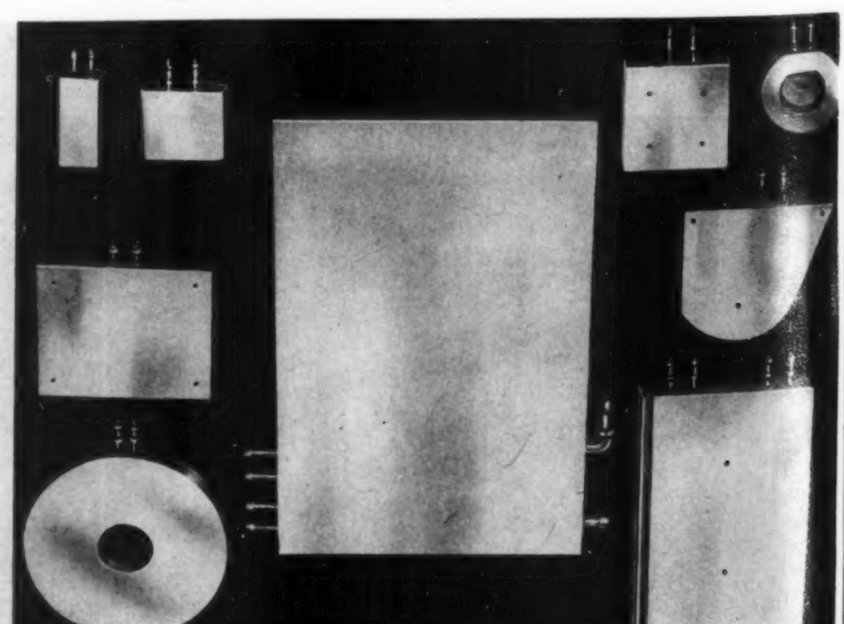
For these reasons, practical usefulness of open elements has been limited to two types of applications; radiant and convection heating. They have been widely used to cost advantage in a variety of ovens both industrial and domestic, and domestic appliances. When properly utilized, the uninsulated and exposed surface of the open element can be of direct advantage. For certain types of gas heating, particularly air, this feature offers a high rate of heat transfer superior to the enclosed units in which heat flow is impeded by the insulation and sheath. Moving air cools the exposed element rapidly, permitting higher heat transfer rates and increasing the efficiency of operation. Attributes of the open element have been utilized to particular advantage in the convection oven design shown in *Fig.* 6.

Radiant heating is another area of application in which the open element can often be used to advantage. Mounted in grooves of parabolic shape in a ceramic form or backed up by a metallic reflector, these elements provide a low-cost high-efficiency heat transfer unit. Again, the elimination of insulation and sheath improves thermal properties and sensitivity of response and control. Best results have been achieved under intermittent operating conditions. For continual operation, especially in industrial applications, the enclosed units have found wider acceptance. Of course, for any radiant heating unit, environmental conditions play an important role; safety or other considerations may render the open element out of the question.

Although open elements are associated with direct heat applications, their thermal properties may also be utilized indirectly. For example, the

Fig. 8—Cast-in tubular units offer a mechanically strong construction and high heat transfer rates





watthour and thermal demand meter in *Fig. 1* employs a pair of heating units, punched from 0.0113-inch nickel-chrome sheet, to indicate peak current loads. The heating units develop a temperature difference proportional to the kilowatt load, expanding a pair of opposed bimetal springs that drive a pusher pointer. This action in turn advances an idle pointer to record the highest reading reached during the month.

Designing Wire Elements: Fundamental considerations in the design of an open element are heat requirements, voltage, rate of heat transfer, operating temperature and space requirements. General rules of practice indicated for wire elements are equally applicable for ribbon types.

The usual design procedure for wire elements is:

1. Calculate heat requirements.
2. Find total resistance required.
3. Select wire size.
4. Determine length of wire required.
5. Select coil size.

Total resistance may be calculated from Ohms law using

$$R = \frac{E^2}{P} \quad \text{.......................................} \quad (11)$$

where R = total resistance in ohms, E = operating voltage and P = energy required in watts. The resistance value obtained from this equation will be the "hot resistance" required at operating temperature. Since most tables of wire resistance are based on room temperature conditions, a corrected or "cold resistance" value will have to be calculated. For the average application, hot resistance will run about 7 to 12 per cent higher than cold resistance in the nickel-chrome alloys, about 4 to 7 per cent more in the aluminum-iron compositions.

Recommended wire sizes and cold ohms for different total wattages for the well established 80-20 nickel-chrome wires are given in TABLE 1. These wire sizes are generally applicable to all nickel-chromium grades and have been established on the basis of recommended usage for cord attached appliances; however, they represent accepted practice

Table 1—Recommended Element Wire Sizes*

Watts at Operating Temperature	Cold Resistance† (ohms)		Wire Size (B & S Gage)	
	110-120v	220-240v	110-120v	220-240v
100	123.5	494.0	26–30	29–33
200	61.8	247.0	25–29	28–32
300	41.2	164.7	24–28	27–31
400	30.9	123.5	22–26	25–29
500	24.7	98.8	20–24	23–27
600	20.6	83.3	19–23	22–26
800	15.4	61.8	18–22	21–25
1000	12.4	49.4	16–20	19–23
1200	10.3	41.2	15–19	18–22
1500	8.2	32.9	12–16	15–19
2000	6.2	24.7	10–14	13–17
2500	4.9	19.8	9–13	12–16
3000	4.1	16.5	8–12	11–15

*80-20 Nickel-chrome wire; †75 F

for most applications and will provide a safe operating temperature. Aluminum-iron wires have a higher resistance value than the nickel-chrome types and can be used in a larger size at the same length to produce an equal total resistance; in the same size, they will allow a shorter element.

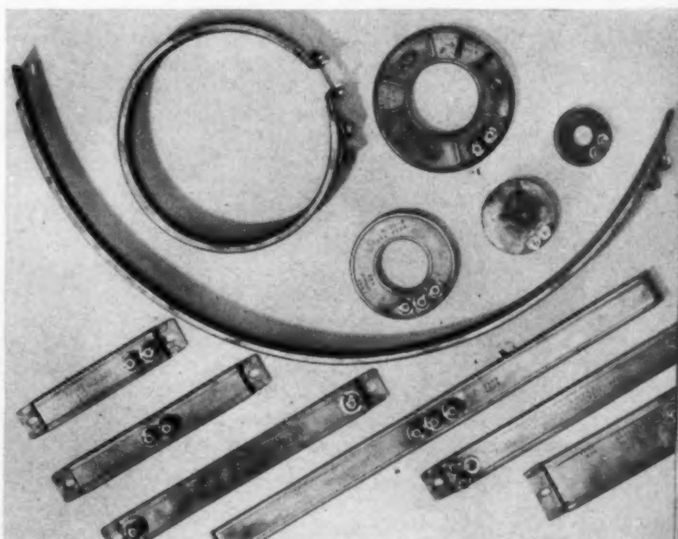
Selection of wire and coil size will be influenced by space requirements and voltage. Larger wire sizes reduce watt loading and operating temperature but, however, increase cost slightly. Coils are usually close wound and stretched to 1½ to 4 times their close-wound length; a value of 2 is desirable. Ratios of coil ID to wire diameter should be kept under 10; higher values will result in a flimsy coil that is difficult to make. Standard tolerances are plus or minus 0.002-inch on diameter and plus or minus 5 to 10 per cent on lengths specified on the basis of resistance.

Elements designed to operate on 230 volts require four times the resistance of those operating at the same wattage at 115 volts. Smaller wire or ribbon sizes will be required for the higher voltage, in many instances making it difficult to design a rugged element, particularly where space is limited.

Choice of wire alloy grade will depend on operating temperature. For a given wattage, element temperature will be determined by the rapidity with which the surrounding medium conducts heat. For example, a coil may operate at 1200 F in open air and at 1650 F when mounted in an insulated enclosure. Permissible watt densities vary considerably with the application, running as high as 50 watts per square inch of exposed element. However, in general practice values of 10 to 20 watts per square inch are usually the goal; for precision work 7 to 10 watts per square inch are considered a practical range.

The following table for heating air in small

Fig. 9—Typical strip heating units showing shapes and terminal arrangements possible. Ring units, a modified version, can be nested for increased wattage



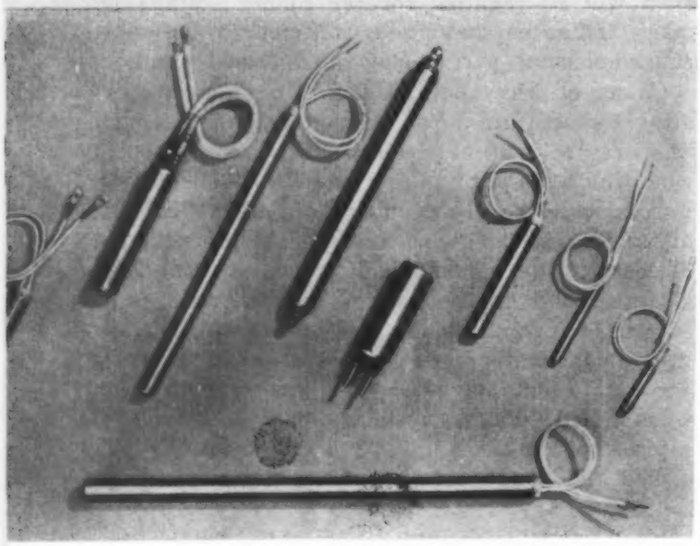
ovens indicates the temperature gradients which are usually necessary and may prove helpful in estimating temperatures.

Air Temperature, F	Element Temperature, F
1000	1500
800	1200
600	1000
400	750

Where practical, actual testing under operating conditions is the best method for determining element operating temperature.

Operating temperature has a significant effect on element life. For example, the life of a 0.025-inch nickel-chromium wire at 1800 F will be approximately 40 times that at 2150 F. The optimum temperature will be the lowest that can be used consistent with the requirements of the heating task.

Fig. 10—Cartridge heating units designed for insertion in a hole to heat surfaces



Installation of Open Elements: Coil installation and mounting may be accomplished in several ways. Elements may be mounted directly on molded-insulator forms or on insulators in metal or wire frames. Embedding in electrical cement often facilitates heat exchange. Allowance must be made for free coil movement to permit expansion and contraction throughout the coil length. If a coil binds in an insulator, sagging may occur and cause a short circuit. Coil supports should be placed at frequent intervals, the spacing being determined by the coil size and installation requirements. If an element must be anchored in any way, tests should be made to determine effects of warpage and "creeping." In all cases, the element should be mounted to provide the easiest path of heat flow into the medium being heated.

Premature failures of elements are usually due to poor design and faulty construction. Terminal failures, in particular, are troublesome. Brass bolts, nuts or washers used to make power connections may oxidize under heat and form an arc which will burn through the element at this point. Although mechanical contacts are sometimes satisfactory, welded, brazed or spot-welded joints out of the heat zone are the most satisfactory. In some instances, pressed or pressure joints will work but should be proved out under the actual operating conditions which will be encountered.

For large wattage installation, three-phase circuits may offer the best solution. Elements are usually designed to provide a balanced load on each phase. Either delta or wye connections may be used. It must be remembered however, that the delta connection provides full line voltage across each phase while in the wye, phase voltage is equal to line voltage divided by 1.732. This difference in voltages can frequently be used to advantage. By employing a suitable switch, the three elements can be readily changed from delta to wye connections

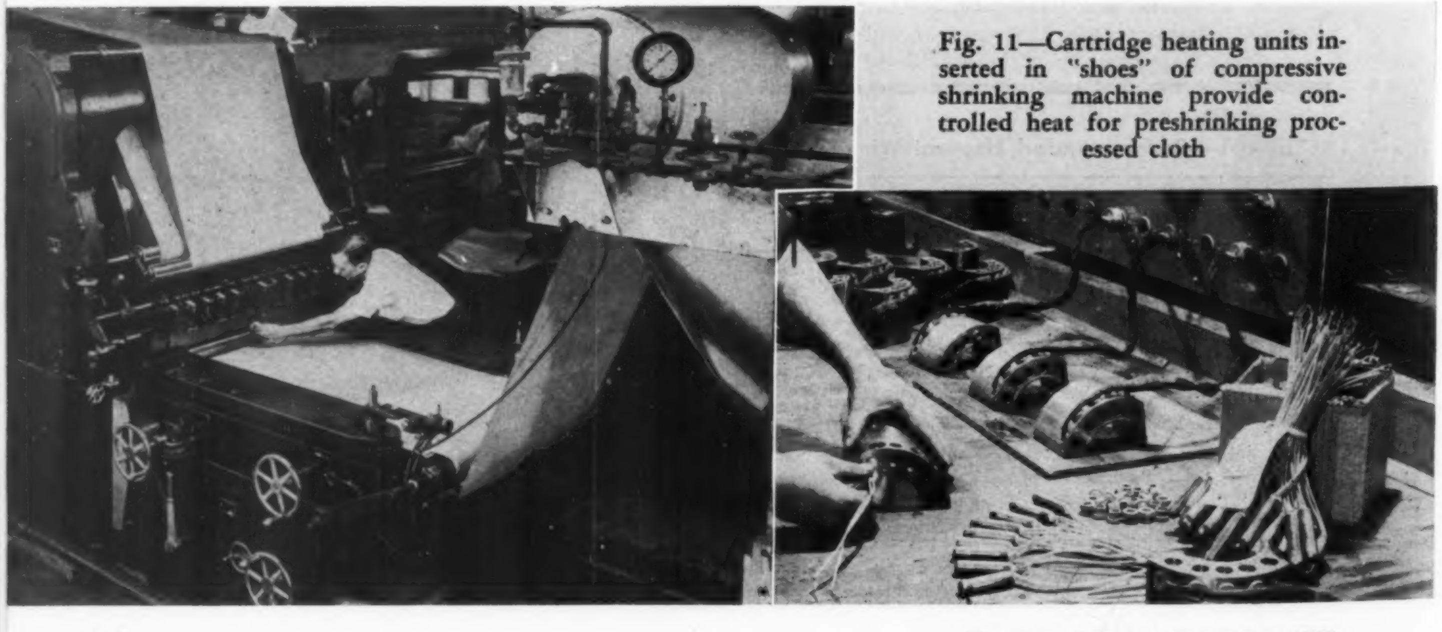


Fig. 11—Cartridge heating units inserted in "shoes" of compressive shrinking machine provide controlled heat for preshrinking processed cloth

to provide two heat controls. Maximum power will be afforded by the delta connection; switching over to wye will reduce input to 1/3. When using such an arrangement, elements should be designed on the basis of the power and voltage supplied through the delta connection.

Enclosed Heating Elements: Metal-sheathed resistance heating units are available in several standard types which may be classified as: (1) Tubular, (2) strip, (3) cartridge, (4) immersion, and (5) fin. Although types and sizes as well as fabrication methods will vary with the manufacturer, most of these units feature essentially the same type of construction; a high-resistance heating element, either coiled wire or ribbon, mounted in a metal sheath and insulated from contact with the sheath by some form of inorganic insulator. The designer must specify the wattage, sheath material, type of terminals and leads, and the effective heating length. Standard voltage ratings are 115 and 230, single and three-phase, ac or dc. Sheath materials include copper, steel, stainless alloys, aluminum, brass, nickel-silver, Monel, Inconel and other alloys.

TUBULAR: These units offer the greatest flexibility in machine applications and can be used for all types of radiant, conduction and convection heating tasks. Typical types are shown in *Fig.* 7. A wide variety of formed shapes is possible and, in addition, these units are particularly suitable for cast-in applications with metals such as gray iron, alloy iron, aluminum, brass, Meehanite, copper and others, *Fig.* 8. Standard available sizes of the tubular type units vary from approximately 1/4 to 3/4-inch in diameter and from 1 to 20 feet in length. Wattage ratings vary from approximately 200 to 7500 watts.

STRIP: Surface heating and natural or forced convection air heating are the primary applications of these thin, flat units, *Fig.* 9, which are useful in restricted spaces. A modified form, the ring unit, offers advantages in certain types of surface heating applications. For use with curved surfaces, these units can be bent either transversely or longitudinally to suit the application. Standard available sizes vary in width from 1 to 6 inches, in thickness from approximately 1/4 to 1/2-inch, and in length from about 7 to 8 feet. A wide variety of terminal and mounting hole arrangements is possible; some types are available with two circuits for providing three wattage combinations. Wattage ratings vary from about 50 to 4500 watts.

CARTRIDGE: Typical units are shown in *Fig.* 10. Designed primarily for insertion in a hole, these units are particularly suitable for contact heating applications where it is necessary to concentrate heat in a confined area, *Fig.* 11. Standard available sizes range from 3/8 to 1⅝-inch diameter and from 1½ to 22 inches in length. Wattage ratings range from 30 to 1800 watts. Either flexible or rigid terminals may be used; both terminals are brought out at one end and the other end can be sealed or open depending on application requirements.

IMMERSION: Liquid heating is the purpose of these specialized units which are placed directly in the fluid medium. Constructions vary widely with the manufacturers but all generally consist

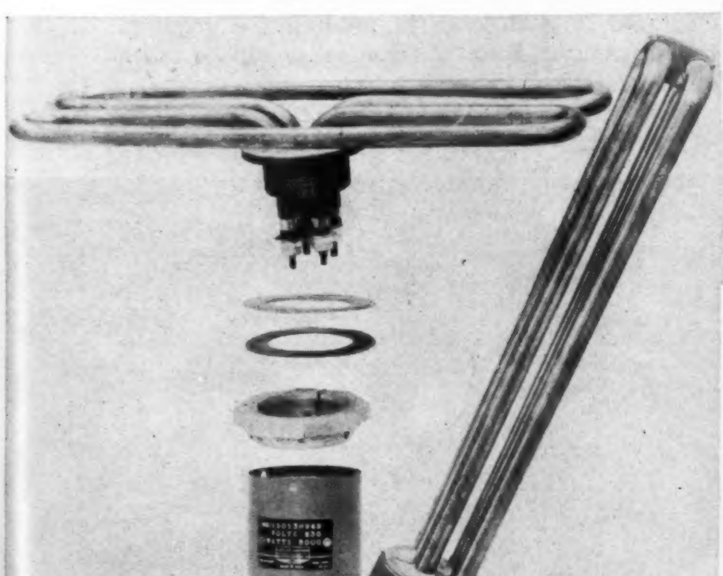
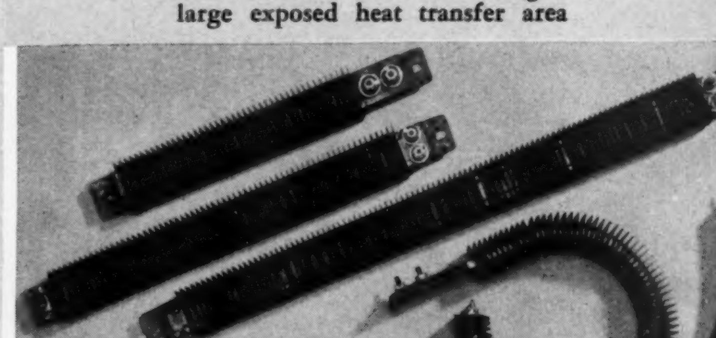


Fig. 12—Left—Immersion heating units for side or bottom mounting offer high heat transfer rates and efficient energy distribution

Fig. 13 — Below — Typical fin heating units for forced convection heating offer large exposed heat transfer area



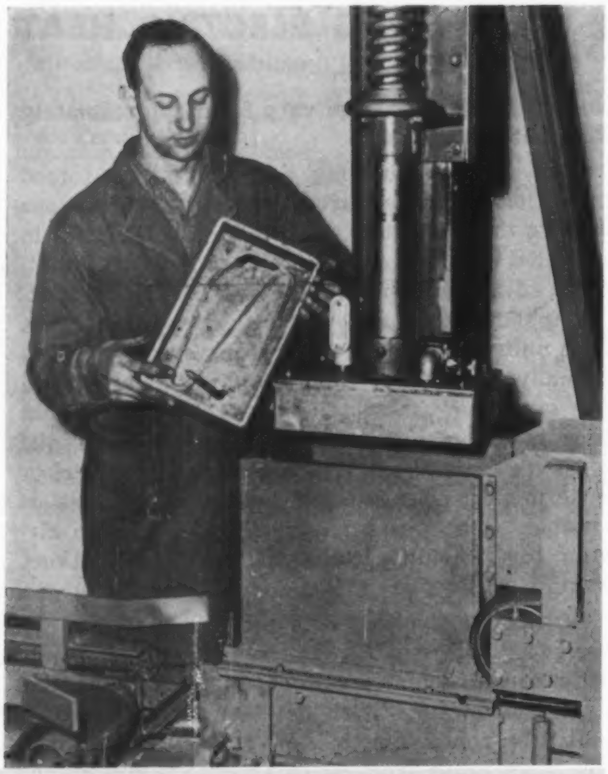


Fig. 14—Above—Efficient heat transfer characteristics of cast-in tubular unit permit accurate temperature control in this dried-fruit compression ram

Fig. 15 — Below — Tubular units clamped in grooves provide high heat transfer rates for surface applications

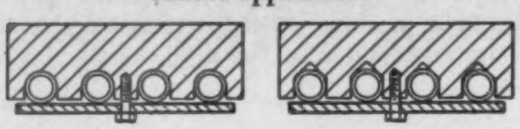


Fig. 16—Below—Compact, efficient design of chamber of injection molding press employs tubular heating units spiralled tightly around exterior surface

of some form of sheathed unit or units mounted to a threaded or flanged header, *Fig.* 12. Sheath material will vary with the liquid being heated and according to its corrosive characteristics. Some of the more widely used combinations are:

Water	Copper, brass
Dowtherm, oil, alkaline solutions	Steel
Acids	Lead, Monel, stainless
Vegetable oil	Nickel-plate copper

Wattage ratings for these units vary from 100 watts, where one element is employed, to 50,000 watts with as many as 18 separate elements.

Modified forms using a cast iron sheath are available for melting soft metals. These are usually designed for "over-the-side" mounting to facilitate removal for maintenance.

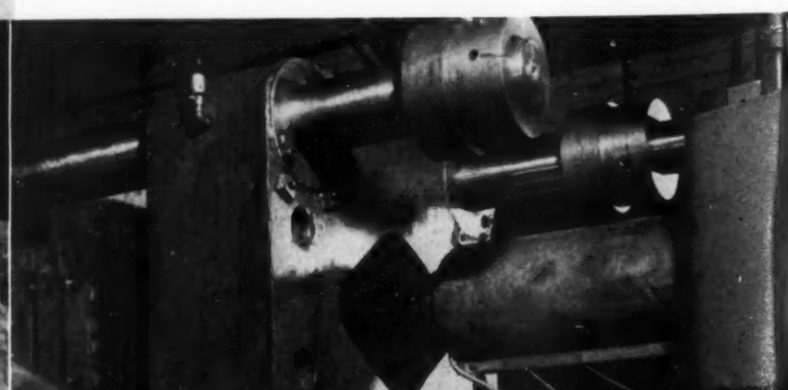
FIN: These are also special-purpose units designed primarily for forced convection heating of gases, particularly air. They usually consist of a tubular or strip unit with either integral or brazed fins, mounted transversely, to provide a large exposed surface. Typical types are shown in *Fig.* 13. Wattage ratings range from 350 to 10,000 watts and lengths vary from approximately 6 inches to 9 feet.

Selecting Enclosed Units: Specification of an enclosed heating unit for a particular application involves certain fundamental considerations including:

(1) Heat transfer media.
(2) Operating temperature.
(3) Rate of heat transfer.

HEAT TRANSFER MEDIA: For the four principal areas of machine applications—surface, air, liquid, metal melting—several methods can usually be employed in each case to produce the required heat capacity. Each of these areas will be taken up separately.

Surface Heating: Three types of enclosed heat-



ing units can be used—tubular, strip and cartridge. The common methods of application are:

(1) Cast-in.
(2) Pressure-clamped in grooves.
(3) Inserted in holes or slots.
(4) Wrapped around surface.
(5) Clamped on surface.
(6) Spaced away from surface.

In general, this listing represents the order of preference in application and is based upon the efficiency with which the contact surface of the heating unit is utilized. Cast-in tubular units offer the best results since full use is made of the entire heating unit surface, *Fig.* 14. The casting effectively increases the surface area of the tubular unit, permitting higher rates of heat transfer than possible with other methods. Heating units of smaller physical size can be used to obtain a compact assembly. In addition, heat losses are decreased and the sheath is protected against corrosion.

Tubular units can also be used to advantage when clamped under pressure in grooves in curved or flat surfaces, *Fig.* 15. Good heat transfer is obtained but surface contact is not as complete as with cast-in types and permissible heat transfer rates are not as high. Variations in sheath and groove diameters will limit the fit that can be obtained.

Another efficient surface heating method is the use of inserted tubular, cartridge or strip units in drilled holes or slots. Here again the fit obtained will limit the heat transfer rates permissible.

For wrapping around surfaces both tubular and

strip heaters can be used. This method has advantages, particularly on round or odd shapes, *Fig.* 16. Strip heaters offer better heat transfer properties since the tubular heaters are essentially in line contact. However, because of adaptability to a wide variety of shapes the tubular heater offers more utility in application, *Fig.* 17.

Strip-heaters, by virtue of their shape, are best suited for clamp-on applications. Tubular heaters can be used but their efficiency is especially limited because of the line contact which results. An interesting application of clamped-on strip heaters is shown in *Fig.* 18, of tubular heating units in *Fig.* 19.

Least desirable of the methods for surface heating is the use of either tubular or strip units spaced away from the surface. Heating will depend on convection and radiation, resulting in a substantial reduction in the possible heat transfer rates that can be obtained.

Liquid Heating: The most effective method of liquid heating is by direct immersion since all heat generated is transferred directly to the liquid. Direct immersion heating is possible through several means: (1) With standard immersion units, (2) by casting-in tubular units, *Fig.* 20, or (3) with hollow tubular sections running through the liquid and carrying replaceable tubular heating units.

When the nature of the liquid or other considerations prevent the use of immersion heaters some

Fig. 17—Accurate temperature control is achieved in the heating unit of this rubber and plastic extruding machine by utilizing wrap-around tubular heating units to meet the odd contour requirements

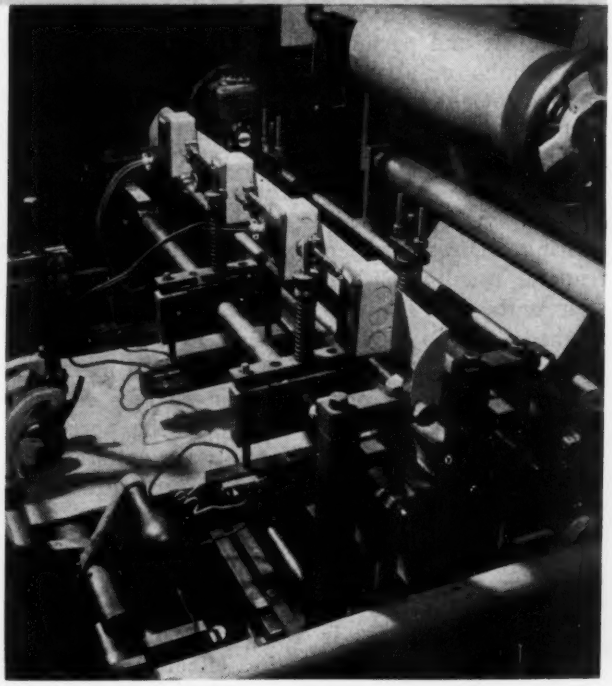


Fig. 18—Clamp-on strip heaters seal paper webs at high rates and under continuous operating conditions in this custom-built envelope machine

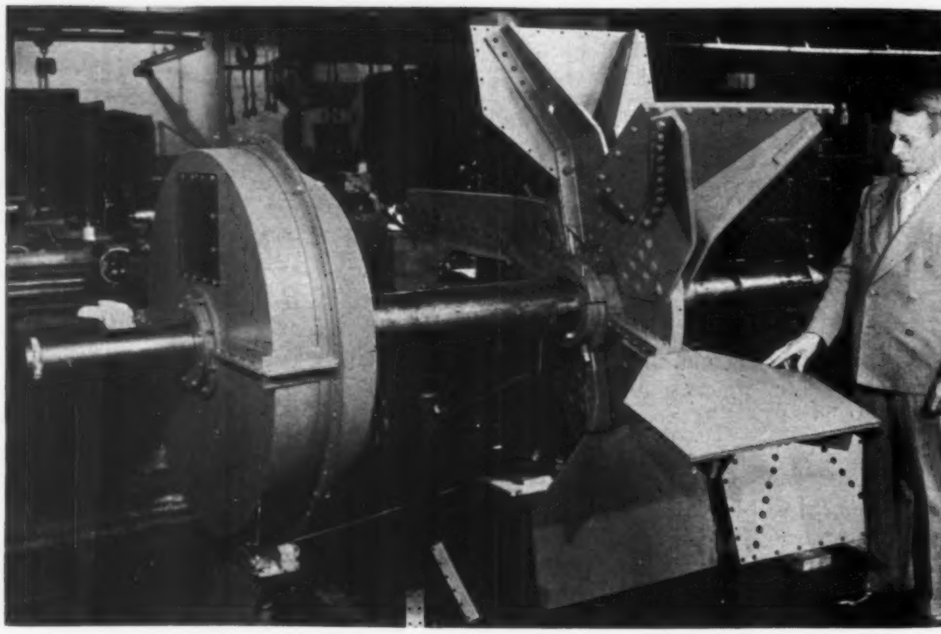


Fig. 19—Left—Blades of sludge drying fan are efficiently heated by tubular units clamped between blade sections

Fig. 20 — Below — Rapid steam generation in this compact insecticide sprayer is obtained with a 1000 watt tubular heating unit, formed in a spiral and cast into brass

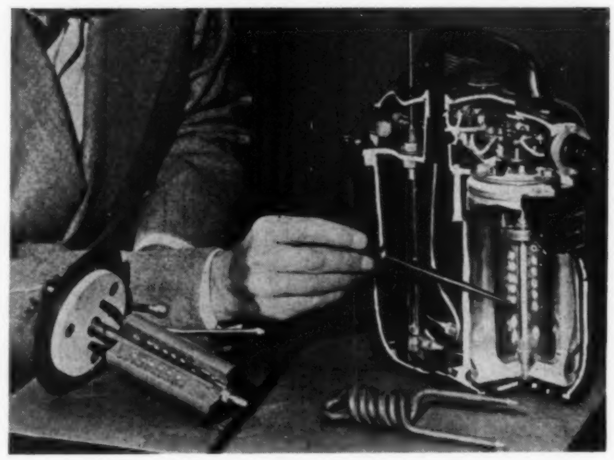
form of indirect heating must be used. This usually can be done through the use of some type of surface heating unit on the container surface or else by a liquid jacket heated by direct immersion. In some instances double-jacketing can be used effectively.

Air Heating: For natural convection heating both strip and tubular heaters offer a range of application. The strip heaters are better suited in most instances for the relatively lower temperature ranges while tubular heaters achieve good results and perform better in the high temperature ranges. In every case, sheath material will have an important influence. Again, the cast-in units also offer possibilities because of their efficient heat dispersal characteristics.

For forced-convection air heating, the fin units offer the most efficient method. Higher heat rate transfer rates are achieved through the exposed fin surfaces which provide a large contact area for the air and permit high-velocity air flow, ***Fig.*** 21. Tubular or strip heating units are also satisfactory in many cases, particularly where air flow is low, temperatures are high and space is restricted.

Metal Melting: Many of the same considerations involved in liquid heating apply equally to metal melting. Direct immersion offers the best heat transfer results. Either the special cast-in immersion units can be used or, in some cases, other methods can be just as effective, ***Fig.*** 22. If direct immersion is not suitable for the molten metal being heated, some form of outer surface heating can usually be employed to obtain the necessary heat capacity. Jacketing is not always satisfactory, particularly where melting temperatures are fairly high.

Radiant Heating: These areas of heat application represent the major ones encountered by the designer. However, one other method of heat transfer, radiant heating, is worthy of mention. The tubular unit, because of its effective reflecting



surface, is eminently suited to this application and has been widely utilized to good advantage. Backed up by a metal reflector, it provides a highly efficient infra-red heater. Today, these radiant units form a specialized group of products and are available in a variety of types and sizes to meet particular application requirements. Design of these units involves a number of technical problems which are beyond the scope of this article.

Operating Temperature: Except for the immersion heaters in which the type of fluid is usually the determining factor, the best type of sheath material is generally established by the operating temperature. Materials are usually selected in accordance with the following table of recommended temperature (F) limits:

Lead	165
Copper	350
Aluminum	625
Brass, Steel	750
Nickel-steel, calorized steel	1000
Chrome steel, stainless, Monel	1200
Incoloy, Inconel	1400-1500

These temperatures are considered the most practical with regard to life and efficiency. Con-

Fig. 21 — Right — Novel popcorn warmer design employs formed fin unit to heat and maintain circulating air at 175F

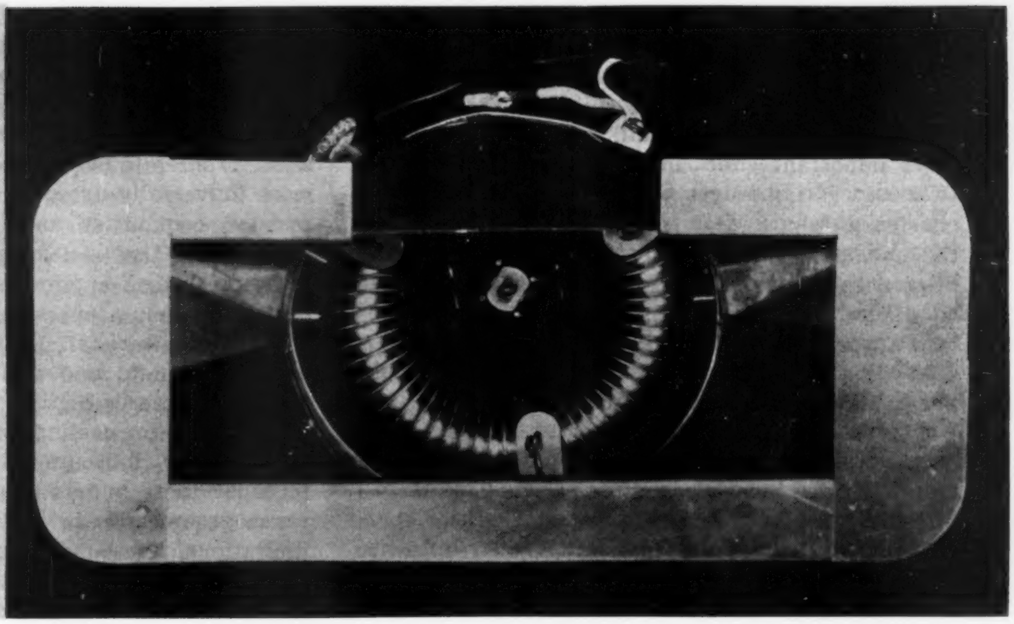
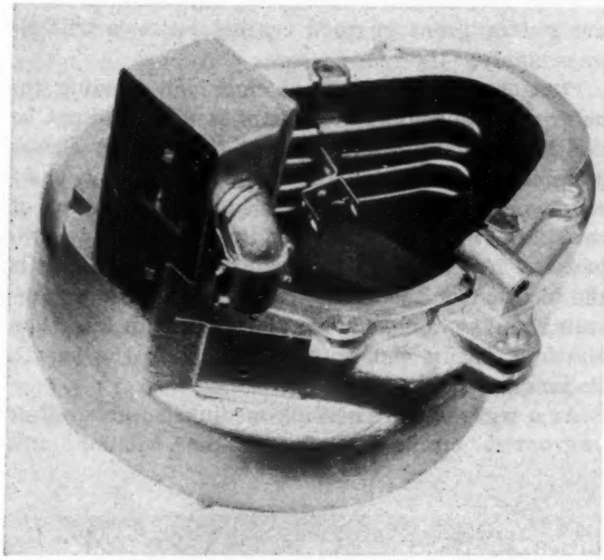


Fig. 22—Below—Tubular units, clamped to walls of monotype pot, provide direct immersion heating capacity of 3000 watts to melt metals and maintain molten liquid temperature



tinuous operation at higher temperatures greatly reduces life and requires special installation.

RATE OF HEAT TRANSFER: A fundamental rule to be followed in heating unit selection is the higher the temperature the lower the watt density (dissipation of energy per square inch). Also, the poorer the thermal conductivity between heating unit and work, the lower will be the watt density. Recommended watt density is a function of the characteristics of the application, the manner of installation and maximum sheath temperature. Exact values for all types are hard to define. For example, 40 watts per square inch is a high value which can only be safely attained if heat transfer is extremly good such as in direct immersion. Cast-in and fin units offer high heat transfer rates because of the increased surfaced obtained which acts to decrease watt density. Efficient heat transfer has a further effect in that it permits units of smaller physical size to be used.

Some of the recommended watt density values for several applications are:

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Sheath	Watt Density	Application
Copper	60	Water immersion
Steel	6-14	Forced air convection
Steel	8	Suspended in free air
Steel	10	Clamped or inserted
Steel	30-40	Cast into aluminum
Stainless steel	25-40	Radiant
Stainless steel	35	Clamped to metal
Inconel	40	Clamped to metal
Inconel	40	Radiant

High watt densities can be used advantageously for certain types of applications where the amount of heat required is limited by the amount of material to be heated. If heating can be accomplished in a relatively short period, dangerous temperature conditions will not develop. However, this method of application should be approached with care.

Watt density can also be affected by the number of heating units. By employing a larger number, lower wattage units can be used and improved heat transfer and performance will result.

Installing Enclosed Heaters: The primary objective of any method of installation is to produce the desired operating temperature with the greatest efficiency. Each of the different methods of heating previously discussed involves certain "rules" which have been established through practice. The following discussion highlights some of the more important details to be observed.

SEALING: Enclosed heating units pose a troublesome sealing problem. Most manufacturers employ a mechanical connection between terminal and coil because of the high heat at the junction. Moisture conditions can cause leakage currents and premature failures. Although complete sealing is impossible because of element expansion and contraction, varying degrees of effectiveness can be usually

achieved to suit the application. Methods range from mica washers for mild conditions to special glass and Mycalex seals for severe conditions. In each case, degree of sealing will affect the cost.

Terminal connections and seals also have another important effect in that they cut down the effective heating length of the unit. However, this shortened length is usually advantageous since the unheated section next to the terminal simplifies external connections. Also, it is usually difficult to utilize full length in most applications and maintain a good connection. If heat were being generated all along the surface, unused sections would overheat and result in failure.

CASTING-IN: Steel is generally employed as the element sheath material because of its all round suitability and comparatively low cost. This sheath material, however, as well as environmental operating conditions of the application, etc., influence the casting metal to be used. For all practical purposes, the free flow path of heat presented by the cast section, regardless of the casting material, will usually make specific differences in heat conductivity relatively unimportant in actual application. Cast metals commonly used are gray iron, aluminum, bronze, brass, and lead. Gray iron is the most universally accepted because of its corrosion resistance qualities and wide range of possible application. Low melting point limits aluminum to low-temperature applications; experience indicates that a maximum operating temperature of 600 F is best for this metal. Bronze is excellent for water immersion units and will usually produce sound castings. Brasses high in zinc content are usually undersirable for casting because of the amalgamating effect which occurs with steel sheaths and tends to cause these metals to penetrate the sheath and cause grounds. Lead may be used, within proper limitations, for applications where the chemical resistance of this material is desirable.

These cast metals only represent some of the more commonly used types. Many other metals are equally suitable and have been successfully used; the general rules of good casting practice will always apply.

FORMING: A good rule of thumb in bending tubular units is that the radius of bend should not be less than twice the diameter of the sheath. Shape possibilities are unlimited, *Fig.* 23, and will usually be governed by the formability characteristics of the sheath material. Cold bending is usually recommended. An important consideration in bending is the location of the terminal junction. For best results, bends should not be located within less than one inch of the junction or damage to the sheath or junction may occur.

As a wrap-around heating medium on grooved or ungrooved circular surfaces, the tubular unit

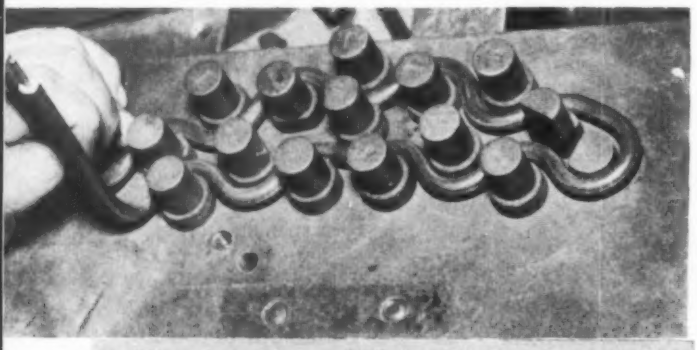
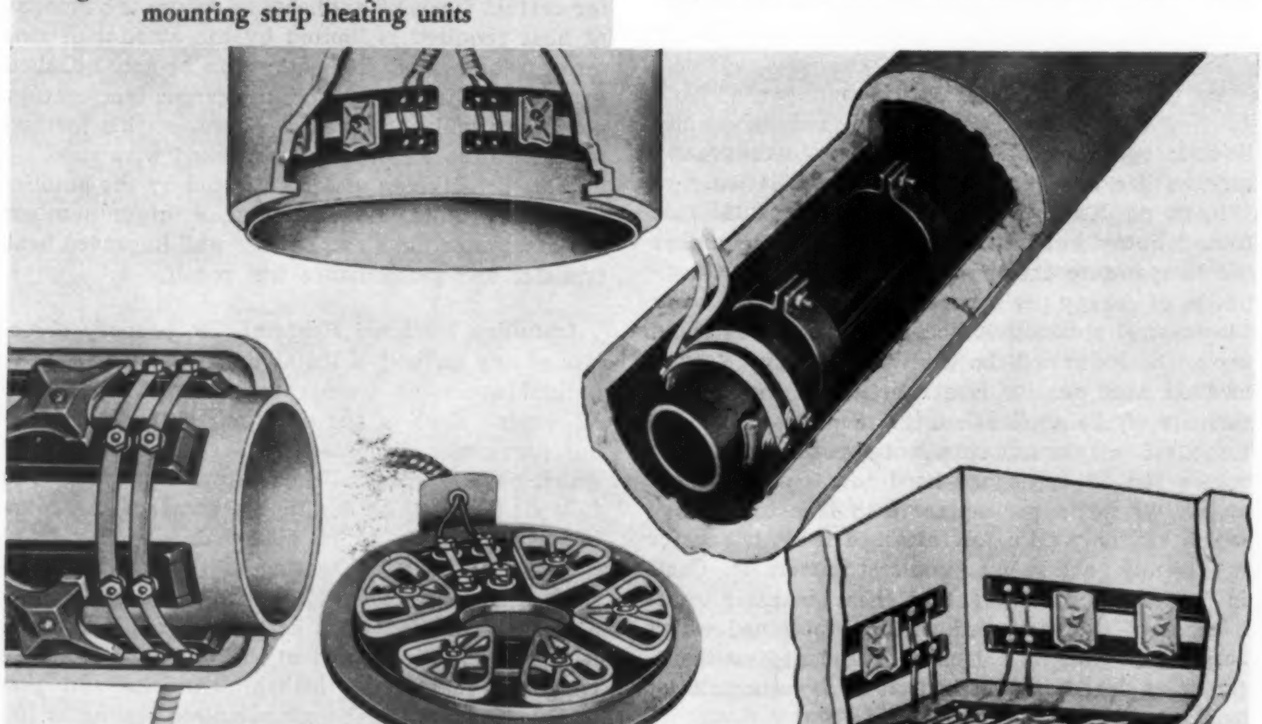


Fig. 23—Above—Heating unit for bottle capping machine illustrates intricate shapes possible with tubular units

Fig. 24 — Below — Clamp-on methods of mounting strip heating units



should be coiled to a diameter slightly less than the heated surface. The unit can then be sprung or threaded onto the surface to obtain a tight fit. Ends should be free to permit expansion and contraction.

CLAMPING: The primary problem in clamp-mounting strip or tubular elements to flat or curved surfaces is to maintain good thermal contact between heating unit and work surface. If the body to be heated is a casting, the mounting surface should be machined when possible. Ends should be free or buckling may occur. The preferred method of mounting is with clamping plates bolted to work on 3 to 4-inch centers. Either studs, screws or welding bolts can be used for anchorage. Typical applications are shown in *Fig.* 24.

Curved strip units for wrapping around tubing offer an additional problem where clamps cannot be bolted to the work surface. One satisfactory method, employs a circular metal clamp with a small expansion gap left between the ends of the strip unit.

INSERTING: Maintenance of surface contact in hole or slot mountings of strip, tubular, and cartridge heaters is hampered by lateral expansion and contraction forces. One satisfactory method is to provide a few thousandths allowance in the slot or hole so that a tight fit is obtained at operating temperature. For the tubular and cartridge units, lubrication with some type of graphite lubricant is often effective; assembly and removal are facilitated and small air spaces are filled assuring good conductivity. A hole leading into the end of the unit which is equal in diameter will facilitate removal of the unit if it should seize. Effect of removal forces can be minimized by placing a heavy end disk on the one end. Where the heated surface is moving, flexible leads provide protection from abrasion or mechanical injury. Several methods of mounting are shown in *Fig.* 25.

IMMERSING: Satisfactory methods of immersion are through the side, or through the bottom of the container. Either threaded, flanged or welded connections are suitable. High watt ranges of immersion heaters necessitate their being covered with liquid throughout the effective heating length. Installation should be close to the container bottom to assure good convection circulation. Uniform heating is thus assured. Space should be provided below heating unit to eliminate danger of air pockets insulating surface and causing hot spots. When heated length extends over 24 inches, supports should be used to prevent sag. In some instances, over the side mountings can be used where removal from liquid may be necessary.

NATURAL CONVECTION MOUNTING: Bolted end mountings are most satisfactory. Maximum air movement is achieved by locating strip or tubular units at the bottom of heated area. For uniform heating, units should be located at sides or, where a high degree of uniformity is desired, on top, bottom, sides and ends. Where explosive vapors are present or damage to heating units is imminent, location should be outside the heating area to assure safety of operation.

FORCED CONVECTION MOUNTING: For maximum heat transfer, fin tubular or strip units should be installed transverse to flow with bolted end mountings. Adjacent rows of units preferably should be staggered. However, such an arrangement does restrict flow to a certain degree. Units should be

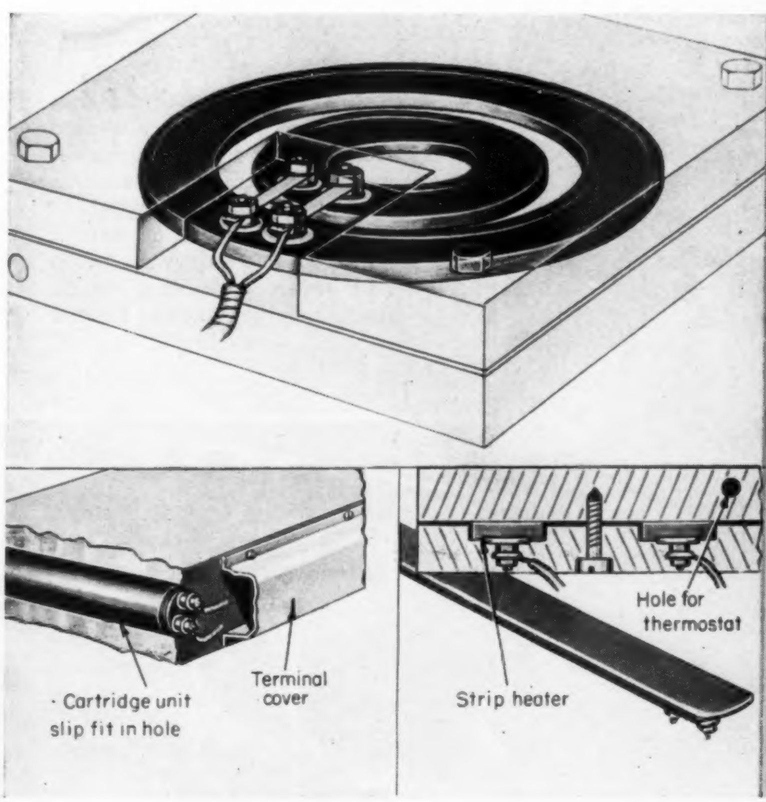


Fig. 25—Left—Methods of installation for heating surfaces with strip, ring and cartridge units

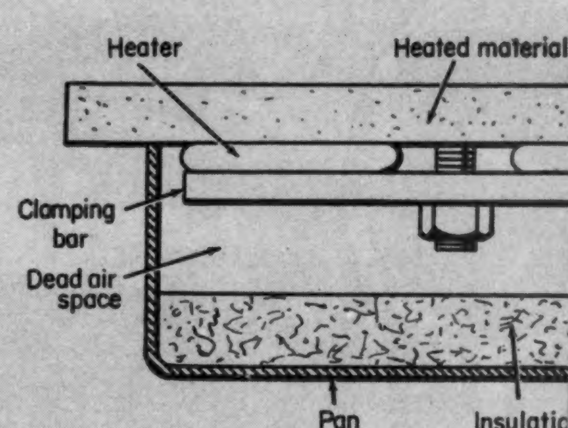


Fig. 26—Dead air space formed by insulating material reduces converted heat losses from backside of heating unit

installed so that they are not energized at rated voltage in still air. Square or rectangular duct installations are desirable where practical; ratio of side lengths should not exceed $1\frac{1}{2}$-to-1.

INSULATING AGAINST HEAT LOSS: In general, efficiency and economy of operation dictate the use of insulation when operating temperatures go above 200 F. For installations involving air heating under continuous operation, insulation at even lower temperatures may be worthwhile.

Contact between insulation and heating unit sheaths should always be avoided. Otherwise, excessive temperatures will build up in localized areas and possibly lead to premature failures. Where possible a sheet metal baffle should be interposed between the sheath and the insulating material.

Dead air space, or the dead air space formed by the different insulating materials, insures against converted heat losses. Radiation losses may be reduced by using a bright reflecting surface behind the heating unit. One satisfactory arrangement is shown in *Fig.* 26.

For liquid heating, side wall and bottom insulation can reduce radiation loss by 10 to 25 per cent. Where practical, insulated covers for exposed liquid surfaces can further reduce radiation and evaporation loss by 50 to 80 per cent.

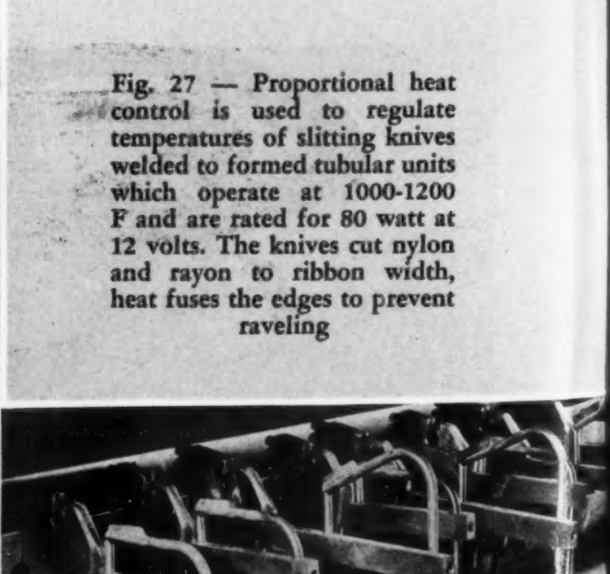
Controlling Temperature: Perhaps one of the most significant advantages of electric heating lies in its controllability. Automatic regulation may be achieved through any of a number of thermostatic devices or manual switching may be used to assure a watt input equal to the energy being used. By using two units having equal ratings with a three-position switch, three separate heat capacities can be obtained. Still further combinations are made possible by three-phase connections, which can be changed from delta to wye connections as described previously.

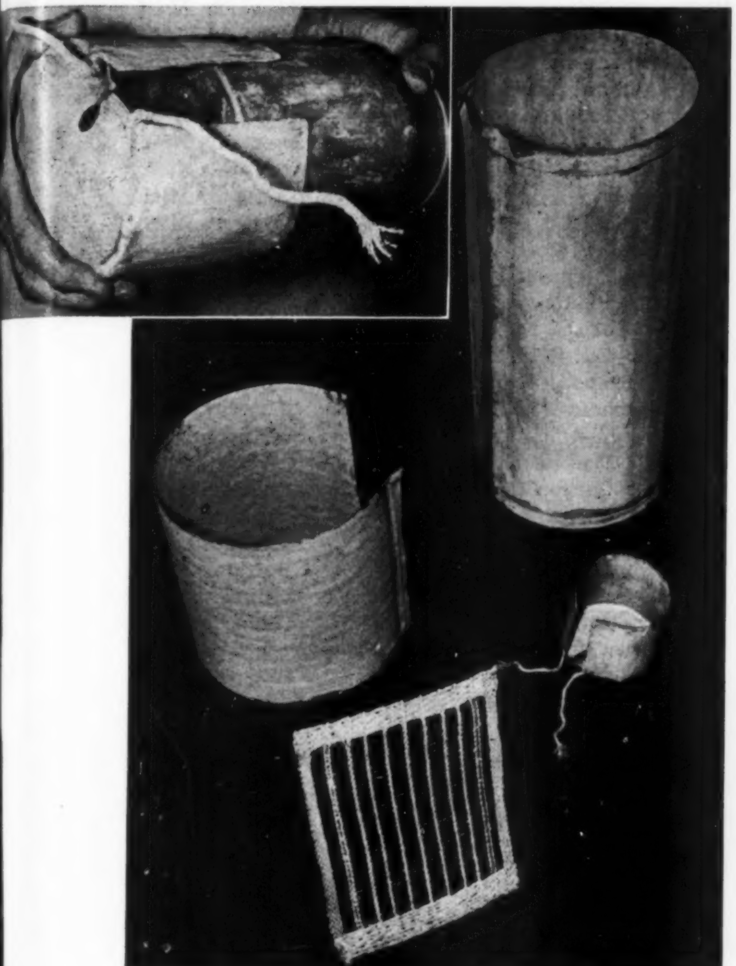
Automatic temperature regulation may be obtained with on-off controls such as the bulb or bimetallic thermostats and pyrometers or with continuously regulating devices which vary the watt input according to the temperature, *Fig.* 27. Accuracy of control desired and the characteristics of the application will be deciding factors. Location of the control should be such that measurements are made in an "average" heat zone.

Recent Developments: Several comparatively new innovations in resistance units offer additional possibilities for specific applications. Most are designed for low watt densities and shape flexibility. In *Fig.* 28, two types are shown, a woven unit constructed of wire and asbestos strands and a laminated type in which resistance wires are sandwiched between two sheets of asbestos paper. Inexpensive and complete with insulation, these units may be readily molded to fit the contours of curved surfaces. Watt densities range from 5 to 10 watts per square inch at ratings of from 6 to 660 volts.

Printed circuit techniques also provide a flexible heat generating unit, *Fig.* 29. These coatings can be bonded to a wide variety of materials ranging from plastics to metals. Uniform surface heating, low element temperatures and efficient energy exchange are achieved at temperatures up to about 400 F. Especially suitable for irregular contours, these units can be used at ratings up to 250 volts ac or dc and provide loadings of 1.5 to 5 watts per square inch.

Fig. 27 — Proportional heat control is used to regulate temperatures of slitting knives welded to formed tubular units which operate at 1000-1200 F and are rated for 80 watt at 12 volts. The knives cut nylon and rayon to ribbon width, heat fuses the edges to prevent raveling





APPLYING ELECTRIC HEAT

Another unique type of element is shown in *Fig.* 30. Identified as film resistors or conductive coatings, these elements consist of a thermosetting resin pigmented with certain discontinuous conductors. They are usually sprayed or painted on a surface and cured. Operating at temperatures up to 350 F and at voltages from 6 to 220 ac or dc, these elements can be applied to metal or nonmetal surfaces and provide watt densities up to 30 watts per square inch. Surfaces up to 50

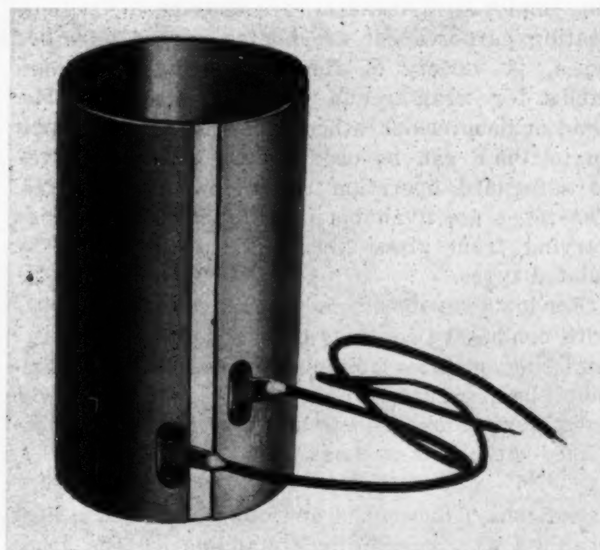
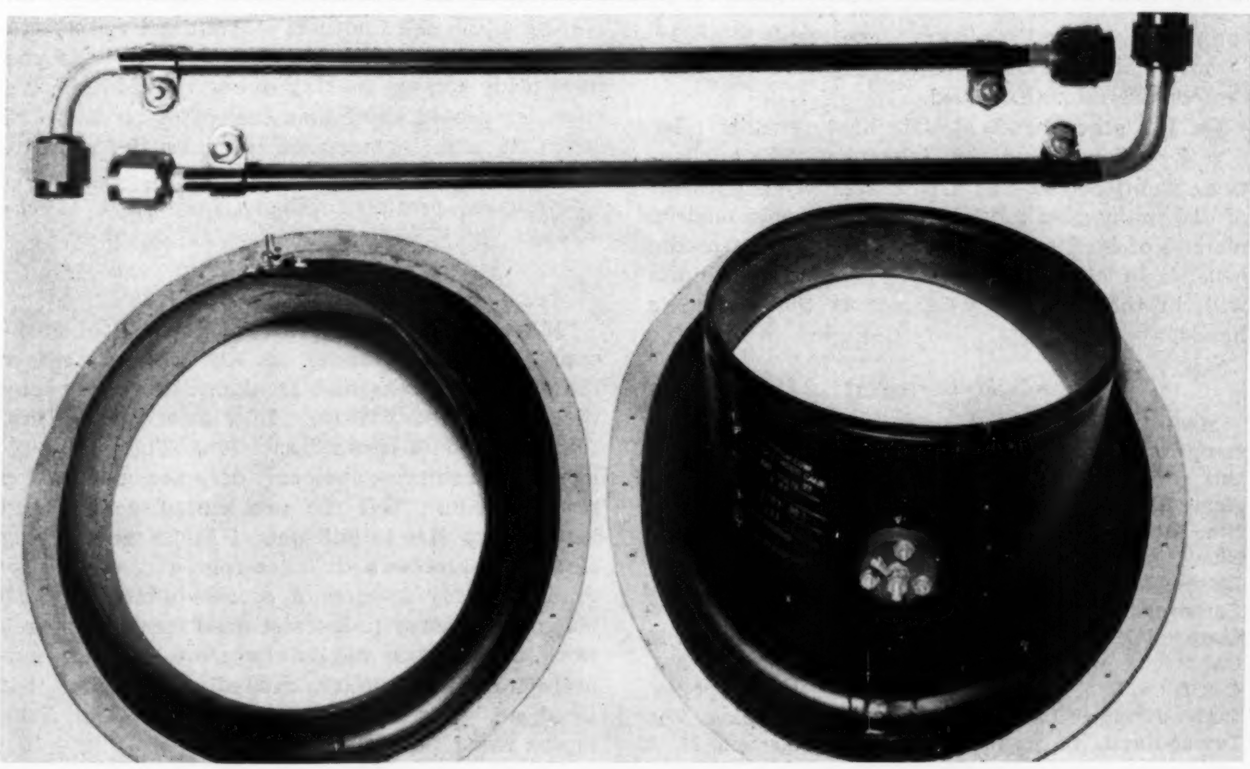


Fig. 28—Above—Woven and laminated heating units for low watt density applications can be readily molded to fit the contours of curved surfaces

Fig. 29—Printed circuit heating elements, bonded to plastic and metal surfaces provide uniform heat at temperatures up to 400 F

Fig. 30—Below—Film resistors, sprayed or painted on surfaces and cured, are used at temperatures up to 350 F



square feet or as small as ½-square inch have been successfully heated. Coating thickness vary from 0.002 to 0.010-inch and several methods of terminal connection can be used.

Flexible serpentine coils of element wire beaded with refractory insulation are useful for surface heating at temperatures up to 1000 F. They can be readily coiled around pipes, cylinders and odd contours. Available in lengths up to 25 feet, their wattage capacities range from 5 to 20 watts per lineal inch. Their flexibility permits them to be tied in knots if necessary.

Several other types, recent and not so recent, also deserve mention. Silicon carbide rods, primarily employed in furnaces, are suitable for radiant heating purposes but are not as rugged as other types. A variety of flexible heating tapes and cables for wrap-around uses are also available. Lead or neoprene-sheathed cables for temperatures up to 165 F can be used around pipes or valves to safeguard operation under low temperatures. The tapes are available in different constructions varying from glass fiber braided to plastic insulated types.

For low temperature radiant heating, glass panels with conductive coatings offer advantages of large radiating surfaces and simple installation. Maximum temperatures which can be reached on the work surface are somewhat lower than can be obtained with other methods.

Summary: Successful application of electric heat requires a thorough understanding of the basic laws of heat transfer as well as of the problems involved in electric heating. Intelligent appraisal of any application must give full recognition to the fact that an electric heating unit is a constant energy device and will develop its rated power output regardless of the surrounding ambient conditions. Dangerous environmental conditions, both for personnnel and equipment, can result if design is not established on a well-rounded knowledge of the considerations involved.

On the other hand, electric heating units offer a real challenge to design ingenuity. The applications mentioned in this article are only suggestive of the many possibilities inherent in this modern method of heat generation. Full realization of the benefits of electric heat in built-in machine units will, in the final analysis, rest in the designer's hands.

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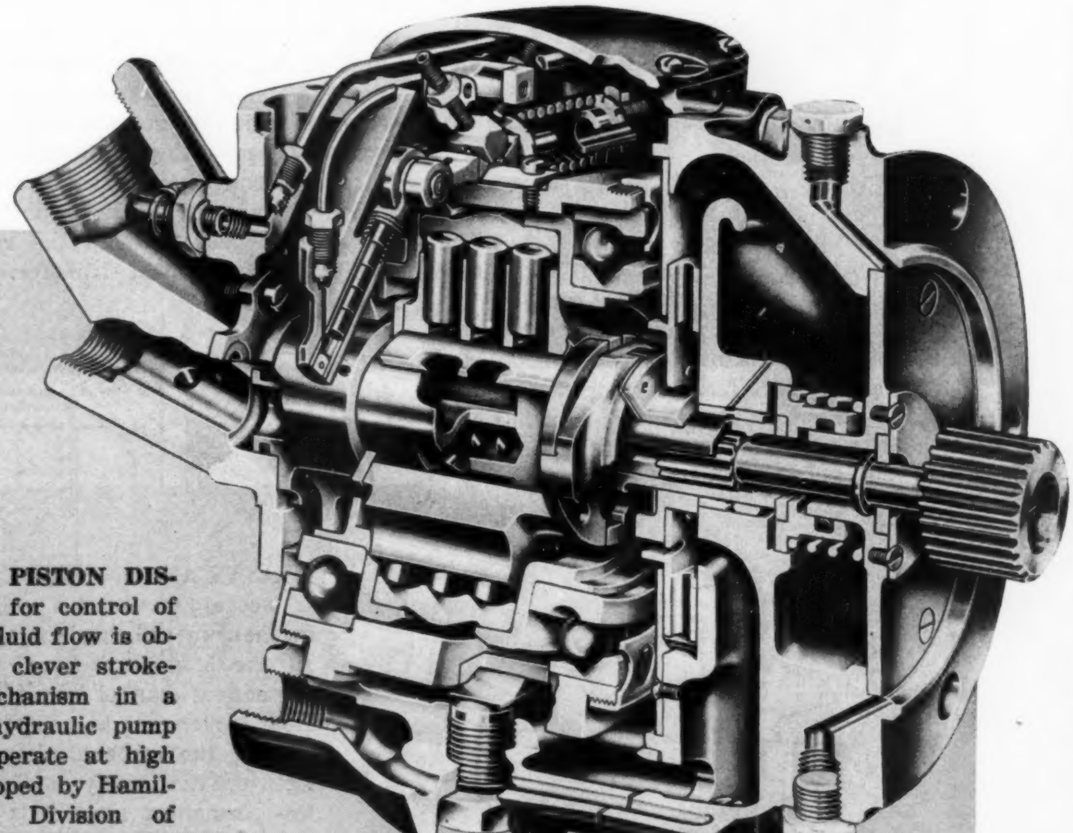
Whirlpool Corp. .... St. Joseph, Mich.

They Say . . .

"Too often the scientist or engineer is regarded more as a diabolical inventor of weapons of destruction than as a patriotic citizen who helps defend his nation. Too often he is seen as a materialistic exponent of a gadget civilization rather than as a vital ingredient in the economy that has made America the wonder and envy of the world. Between us, we must bend every effort toward replacing these false notions of your activities with a truer regard for the manifold contributions you have made and are making to our way of life. We must let people know that the major objective of scientists and engineers is to make the world a better place in which to live and work."—JOHN T. RETTALIATA, *president, Illinois Institute of Technology.*

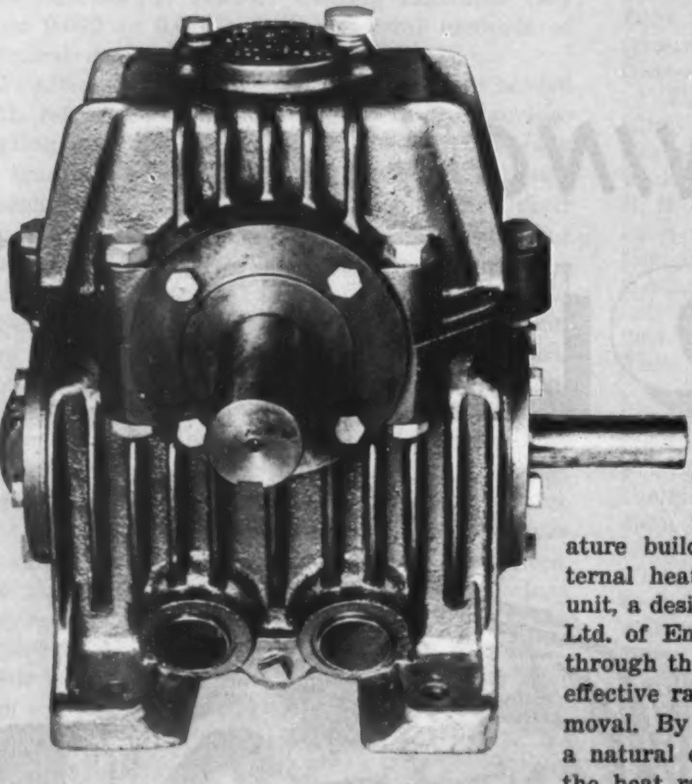
"It has been said that the employment of mediocre research personnel at low salaries, rather than top-flight personnel at high salaries is 'penny wise and pound foolish.' Like most generalities, this one needs a closer inspection. The mere paying of high salaries obviously does not create a top research man. But the problem of getting and keeping top men is still one of the most harrying of all the problems that confront a research program. In my opinion, a specific effort must be made by industry to interest good men in research work, and to train and develop more research engineers for the constantly expanding programs that lie ahead."—LESLIE R. GROVES, *vice president, Remington Rand Inc.*

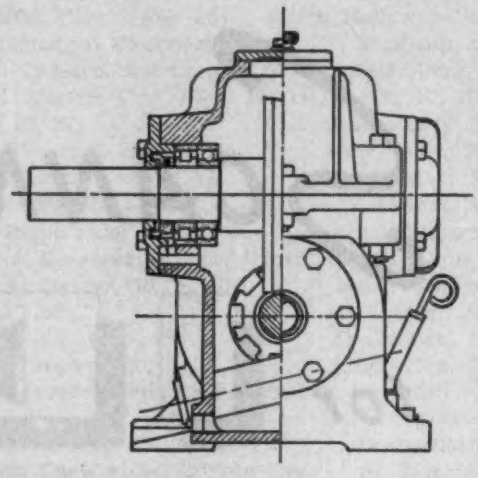
SCANNING the field for IDEAS



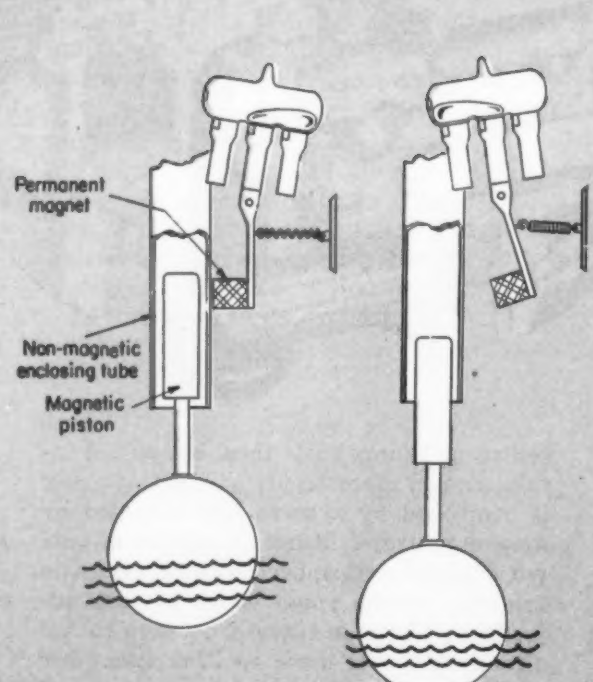
VARIABLE PISTON DISPLACEMENT for control of pressure and fluid flow is obtained with a clever stroke-adjusting mechanism in a radial-piston hydraulic pump designed to operate at high speeds. Developed by Hamilton Standard Division of United Aircraft Corp., the Hydromatic pump employs a pivoting track ring to regulate the pumping action of 21 radial pistons reciprocating in a rotor revolving about a fixed center at speeds up to 9000 rpm.

In operation, the piston heads bear against the inner surface of the track ring mounted in a ball bearing gimbal which pivots about a knife edge. Pumping cycle is obtained by the eccentricity of the track ring. The pistons are forced toward the center of rotation on the pumping stroke, centrifugal force holds them out on the intake stroke. Eccentricity of the track ring is controlled by a servo unit actuated by system pressure. Infinite variation of output characteristics over a wide range is achieved by the pump which permits adjustment of piston travel from zero to full stroke at speeds down to 3750 rpm. For special applications, other pressure control mechanisms can also be employed.



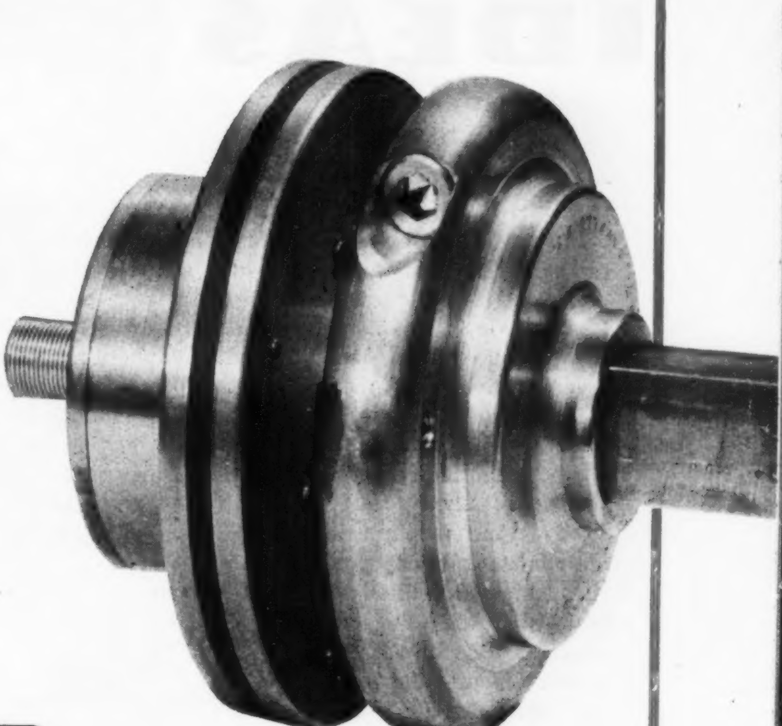


CONVECTION TUBE COOLING of enclosed gear drives reduces temperature buildup and simplifies dissipation of internal heat. In the Convecool worm reduction unit, a design developed by Schofield and Samson Ltd. of England, two nonferrous tubes passing through the oil supply substantially increase the effective radiating surface available for heat removal. By mounting the tubes at a slight angle, a natural convection circulation is produced by the heat radiated at the tubing inner surfaces. The cooling tubes pass through the reservoir of oil, assuring high rates of heat transfer.

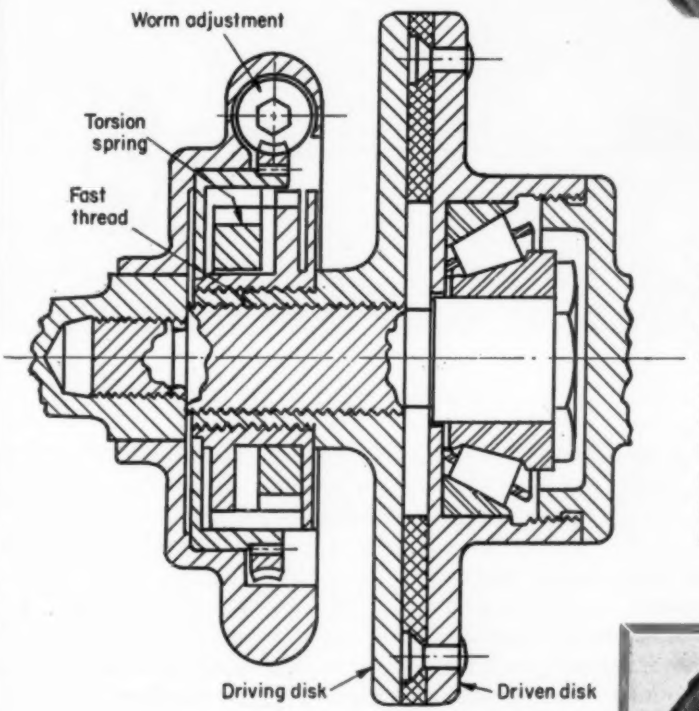


ACCURATE CONTROL of liquid levels is accomplished with a novel switching mechanism utilizing a magnetic "linkage." In the Magnetrol liquid level control, a permanent magnet is employed to transmit float movements to a mercury switch, eliminating the mechanical or electrical linkages conventionally employed. Position of the permanent magnet is controlled by a magnetic piston attached to the float. At normal operating level, the piston holds the magnet firmly against the side of a nonmagnetic cylinder, establishing one circuit of the mercury switch. As the liquid level recedes, the piston is carried down with the float, releasing the permanent magnet from the influence of the magnetic field. Gravity and spring action cause the magnet to swing-out, reversing the circuit of the mercury switch. Moving parts are kept to a minimum by the design which reduces maintenance problems and assures positive control response.

IDEAS



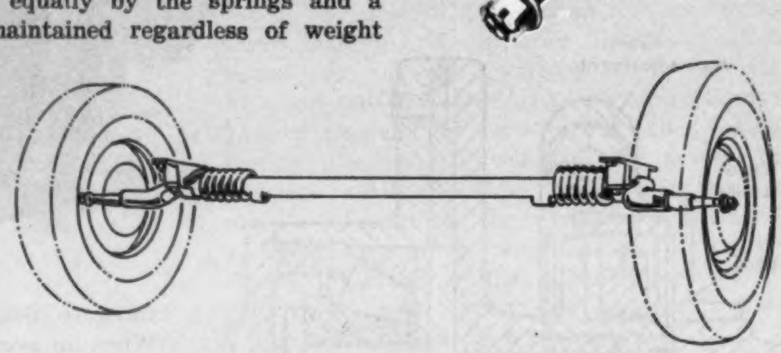
CONSTANT-TORQUE clutch design developed by Buffalo Machinery Co. automatically compensates for friction variations and can be preset to disconnect at a prescribed load. Torque transmission accuracies of ± 5 per cent are assured by a unique construction which maintains a constant friction force at the contact surfaces. Successful applications of the design have been made in torque capacities varying from 5 to 1500 inch-pounds.



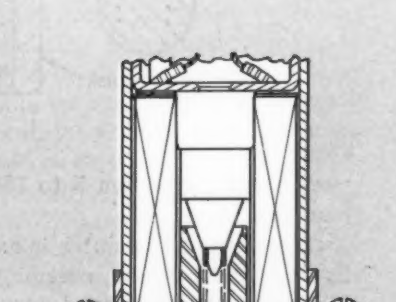
Operation of the clutch is controlled by a torsion spring assembly which applies a predetermined torque to a friction disk mounted to a fast screw on the input shaft. This spring action drives the disk into engagement with a mating disk on the driven member, the thrust developed being a function of the applied spring torque. Troublesome effects of variations in the coefficient of friction caused by changing surface or load conditions are thus eliminated by the design which automatically adjusts the friction force to deliver a constant torque. When an overload occurs, the driving disk overcomes the action of the torsion spring and backs up on its thread, permitting the clutch to slip until loads return to normal. Adjustment of the torsion spring is accomplished externally through a worm and gear arrangement, without requiring disassembly.

IDEAS

TORSIONAL LOAD CONTROL in a novel spring suspension for trailer axles prevents unbalanced force reactions on mounted frames and provides straight, level tracking of towed equipment without equalizing bars. The "Level-Load" axle, an American Steel Foundries-Linco development, utilizes a one-piece forged construction and a spring leverage action to absorb loads and cushion road shocks; torsion spring mountings control the movement of two offset wheel spindles which function as crankarms and deflect identically through a limited arc under load. Axle loads are always shared equally by the springs and a level frame position is maintained regardless of weight distribution. In addition, the design simplifies installation; reduces necessary frame clearance; eliminates surging, rocking and swaying; and facilitates installation and disassembly.



FLOATING SHEAR SEAL design minimizes flow losses and simplifies assembly and maintenance problems in solenoid valves developed by Valcor Engineering Corp. The design employs a carbon-graphite disk, loosely inserted in a solenoid actuated plunger, to act as the sealing element and control flow in two directions through the valve. Fluid pressure seals the disk securely against mating faces on the valve seats, eliminating the need for spring loading or auxiliary sealing means. Valve life and effectiveness are increased by an inherent self-cleaning and self-lapping action during operation. Precision fits are eliminated by insert type construction, facilitating repair and replacement. In addition, pressure drop through the valve is kept to a minimum by a straight-through path of flow.



How to Prevent OIL FILM WHIRL in Journal Bearings

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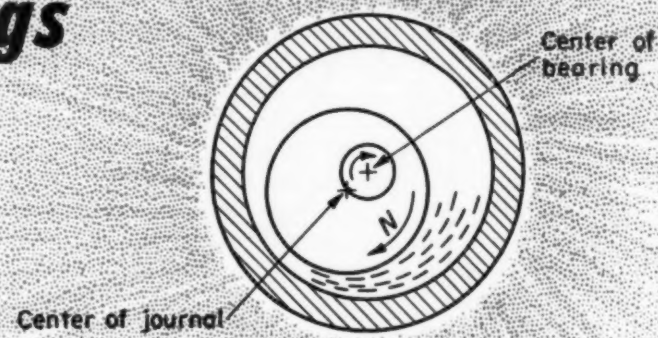


Fig. 1—Oil film whirl, essentially rotation of the journal center with respect to the center of the bearing

OIL FILM whirl is an elusive problem which can sometimes require the expenditure of quite a bit of time, effort and money in its solution. A rational approach to its minimization, however, can eliminate some of the costly effort when encountering the problem for the first time. The procedures presented in this article for solving the problem of oil film whirl may also serve to forewarn the designer of potential whirl and indicate methods for steering the bearing design away from danger areas.

Oil film whirl of plain journal bearings is a condition in which the center of the journal travels in an enclosed path about the center of the bearing in the direction of journal rotation, *Fig.* 1. When this condition exists, frequency of whirl is normally something less than one-half journal rotational frequency and occurs at a definite frequency—not erratically. Further, oil film whirl seems to occur largely in lightly loaded journal bearings. However, it does not follow that all lightly loaded journal bearings will be afflicted with oil film whirl. Some outside exciting force seems required to start this whirling action.

Types of Journal Instability: Other forms of journal instability sometimes produce effects similar to oil film whirl, worth mentioning as a further attempt to isolate oil film whirl, are:

SHAFT DEFORMATION: If a shaft at rest, supported by clearance journal bearings, is heavy enough to have substantial elastic deformation, the center of the journal will not necessarily coincide with the center of the bearing. Rotation of this shaft occurs about its elastic axis rather than a line through the centers of the bearings. Thus, if the shaft passes through a critical speed, the center of the journal may rotate in a closed path about the bearing center but at the same frequency as the journal. This is not oil film whirl.

HYSTERESIS: Newkirk[1] has shown that a journal can whip because of internal friction or hysteresis in the material of the rotating element. Internal friction of a material subjected to a cyclically varying load produces a condition in which stress is not zero when strain is zero—also the converse. A very clear exposition of the result of this hysteresis is given by Shaw and Macks[2]. Frequency of this type of shaft whirling will usually be very near the natural frequency of the rotor or shaft. However, the shaft can whirl at all rotative speeds above the critical speed if sufficient hysteresis is present in the material. Thus, this type of whirl is different from the oil film which is discussed here.

DRY FRICTION WHIRL: Still another form of shaft whirl can exist either as the journal starts initial rotation, or under action of heavy loading. This condition—dry friction whirls—takes place when metal-to-metal contact occurs between the journal

1. References are tabulated at end of article.

and the bearing, and the journal rolls on the bearing. Frequency of this type of whirl can be vastly larger than the rotational frequency of the shaft. If such a condition exists for sustained periods of time (it doesn't have to be minutes), enough heat energy can be released to cause complete bearing failure. As a further distinction from oil film whirl, the direction of dry friction whirl is opposite from that of shaft rotation.

Causes of Oil Film Whirl: Although oil film whirl actuallly develops within a plain bearing, its symptoms and bad effects are experienced in the resonant response of the rotor, which usually cannot be tolerated. For example, a rotor carrying a gear supported on plain bearings may vibrate and clatter so badly during oil film whirl that an immediate change of rotational speed is required. Yet being sure that oil film whirl is the cause of the trouble is far from easy.

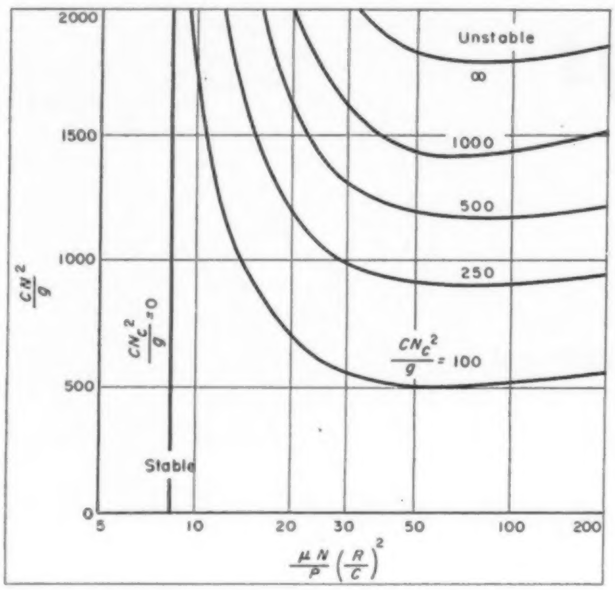


Fig. 2—Stability chart for determining zones of stability and instability for 120-degree centrally loaded clearance bearings with a length-diameter ratio of 1

In 1925, Newkirk[3] established that oil film whirl results from the action of thick film lubrication in a journal bearing. This he did by the novel, but not obvious, device of stopping flow of oil to a journal bearing undergoing whirl. When the oil flow was stopped, whirl ceased.

Although a lightly loaded high-speed journal bearing may seem to develop oil film whirl without any external exciting force, the exciting force appears to be necessary. As will be shown later, a thick-film lubricated bearing has potential instability which can sustain oil film whirl once the center of the journal beings vibrating. Yet the same bearing, if not disturbed, may operate with the centers of the journal and bearing at a fixed relative position, and no oil film whirl develops.

Studies[4,5] of the action of thick film journal bearings have established that potential instability capable of sustaining oil film whirl exists. These same investigations show that maximum frequency of oil film whirl in an unloaded ideal journal bearing (no end leakage) is one-half the journal rotational frequency. A concise treatment of this point is presented by Shaw and Macks,[2] who prove that the general hydrodynamic equations for a dynamically loaded sleeve bearing of infinite length (neglect-

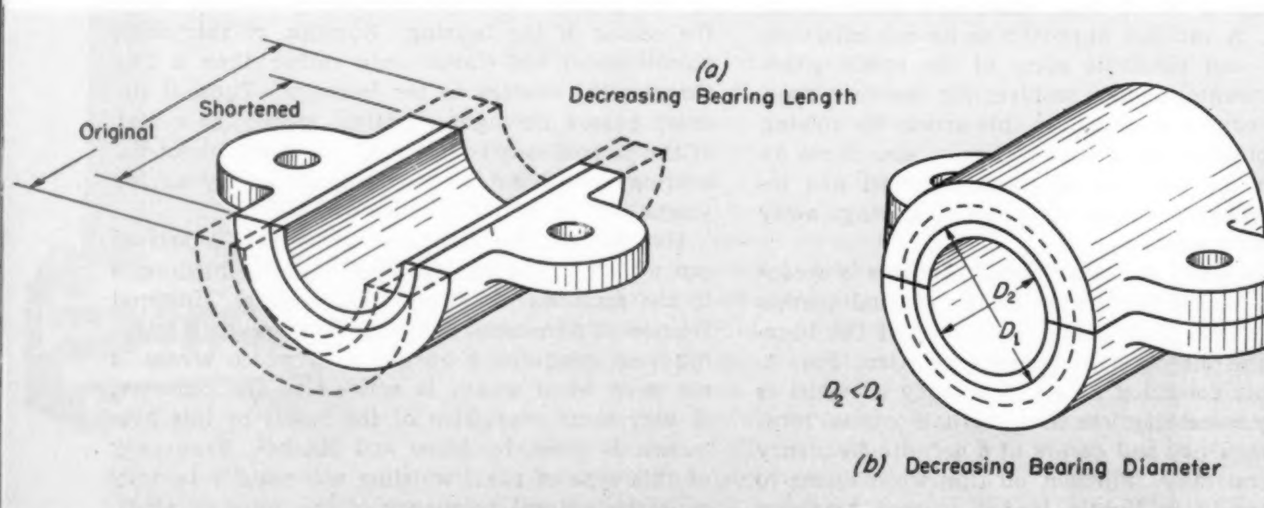


Fig. 3—Methods of eliminating oil film whirl by reducing load-carrying area

ing friction forces) can be satisfied with either of the following conditions:

1. Center of the journal stationary with respect to the center of the bearing.
2. Center of the journal rotating with respect to the center of the bearing at a maximum frequency of one-half shaft rotational frequency.

An outside force is required to start oil film whirl. However, once it begins, it will sustain itself by action of the oil film alone unless there is sufficient damping within the bearing to bring the journal center to an equilibrium condition. Further, if the lowest resonant rotor frequency is less than one-half shaft rotational frequency, oil film whirl may possibly be sustained from this continuous exciting force.

In a real journal bearing (with end leakage and some load), frequency of oil film whirl is generally considered less than one half the shaft rotational frequency. Cases have been reported in which the frequency of oil film whirl in actual bearings was greater than one-half shaft rotational frequency.[6] But in view of the limiting frequency established from fundamental considerations of ideal journal bearings, what was called oil film whirl may not have been that alone. It is not too difficult to mistake oil film whirl for some other form of shaft whipping.

Practical Remedies: Theoretical considerations of the remedies for oil film whirl are covered in the Appendix. Practically, the problem of oil film whirl in a journal bearing may arise either during the design stage or after the bearing is in service. In the former case, the emphasis is on prevention; the latter requires remedial action. The design

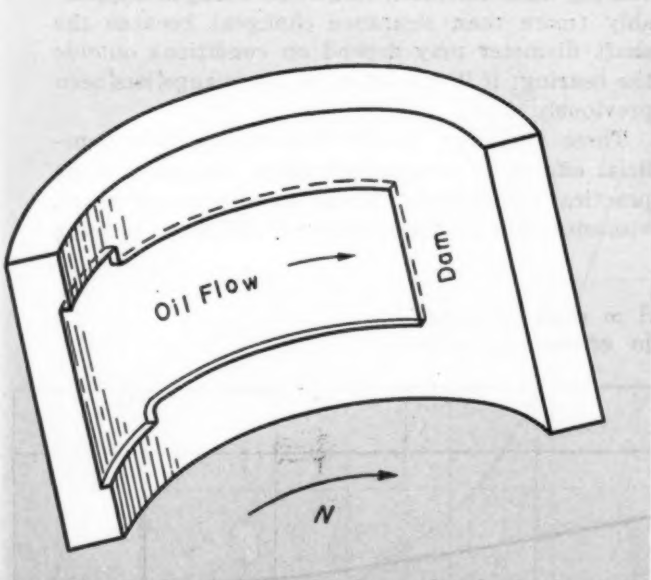


Fig. 4—Bearing cap for increasing bearing pressure by means of special grooving

engineer has to be able to prevent as well as remedy oil film whirl. These two approaches can be considered separately.

Preventing Oil Film Whirl During Design: In the design stage, it is desirable to recognize whether a journal bearing may be susceptible to oil film whirl in order that preventive action can be taken. How can this be done?

An excellent paper by Hagg and Warner[7] presents a criterion for shaft stability which now has the increased validity of an extensive experimental program. Previously this criterion had largely mathematical basis.

Basically, these authors have established that zones of stability and instability, *Fig.* 2, can be outlined for particular bearing configurations on charts having dimensionless abscissas $(\mu N/P)(R/C)^2$ and ordinates CN^2/g by lines representing various values of the rotor parameter $CN_c{}^2/g$ (see Nomenclature).

In practice, $CN_c{}^2/g$ is computed for the shaft to determine the particular rotor parameter line. Evaluation of CN^2/g and $(\mu N/P)(R/C)^2$ for the actual bearing then establishes a point on the chart. If this point is below the line, the bearing design is in a stable region. Several other stability charts are also presented for other L/D ratios and load carrying angles. Another stability criterion is in terms of eccentricity ratio and related attitude angle.[8]

What is the lowest resonant rotor frequency, and what is the shaft rotational frequency? Both theory and practice suggest that the lowest resonant shaft frequency should be more than half (say 0.6) of the rotational frequency. This will not always prevent oil film whirl, but it will always minimize the disturbance. Rotor stiffness should be adjusted to give a suitable relationship between lowest resonant frequency and rotational frequency. However candor requires mentioning that large central-station generators often have rotor critical speeds less than one-half running speed with rare incidence

Nomenclature

C = Bearing clearance, inches
D = Bearing diameter, inches
E = Eccentricity ratio, e/C
e = Bearing eccentricity, inches
g = Gravitational acceleration, ft per second^2
h_0 = Oil film thickness, inches
L = Length of bearing, inches
N = Shaft speed, rpm
N_c = Lowest critical rotor speed, rpm
N_w = Oil film whirl frequency, rpm
P = Bearing pressure, psi
R = Radius of bearing, inches
Z = Absolute oil viscosity, centipoise
μ = Absolute oil viscosity, reyns (lb-second/in.^{2})

of oil film whirl.

Is the bearing to operate at high speed and light load? These are relative terms; however, actual experience is revealing if not particularly significant. Shaft surface speeds above 100 feet per second have been considered high speed, and oil film whirl frequently has been observed above this value. However, surface speed really is not a suitable criterion. Neither is shaft speed alone. But shaft speed and bearing pressure considered together are of interest. Newkirk[9] indicates some results where a 2-inch diameter shaft experienced oil film whirl at 22,000 rpm and 133 psi bearing pressure. Yet this bearing, after being shortened operated at 30,000 rpm at a bearing pressure of 266 psi without oil film whirl. These numerical values are not to be given undue significance, for a bearing with much lower pressures (or even no bearing pressure) might whirl at much lower speeds, particularly vertical bearings with small radial loads.

Whenever the design engineer gets out of a speed and bearing pressure range known to be safe, he may have to hedge against possible oil film whirl difficulties. One of the surest ways is to design in such a manner that the bearing pressure can be increased if this is needed. Frequently, this means being able to shorten the bearing.

Redesigning to Eliminate Whirl: Other shaft disturbances have some similarity to oil film whirl, so in an actual practical situation it is absolutely necessary to be sure that oil film whirl exists. If true oil film whirl is present, a reed-type tachometer will give two readings; one at the shaft speed and the other at less than half the shaft rotational speed. Such information is usually good evidence. Fortunately, a number of practical remedies for such designs exist which are worthy of attention.

Increasing bearing oil film pressure is a way to eliminate oil film whirl. Only two methods of increasing bearing pressure will be considered. One involves reducing the load-carrying area of the bearing; the other involves loading the top of the bearing to increase downward pressure on the journal.

The load carrying area of a bearing can be reduced by shortening its length, decreasing its diameter, or increasing the size of the chamfer at the bearing split, *Fig.* 3. Frequently, the shaft diameter cannot be changed because it may already be established from strength or critical speed conditions. Also changing the shaft size may cost too much.

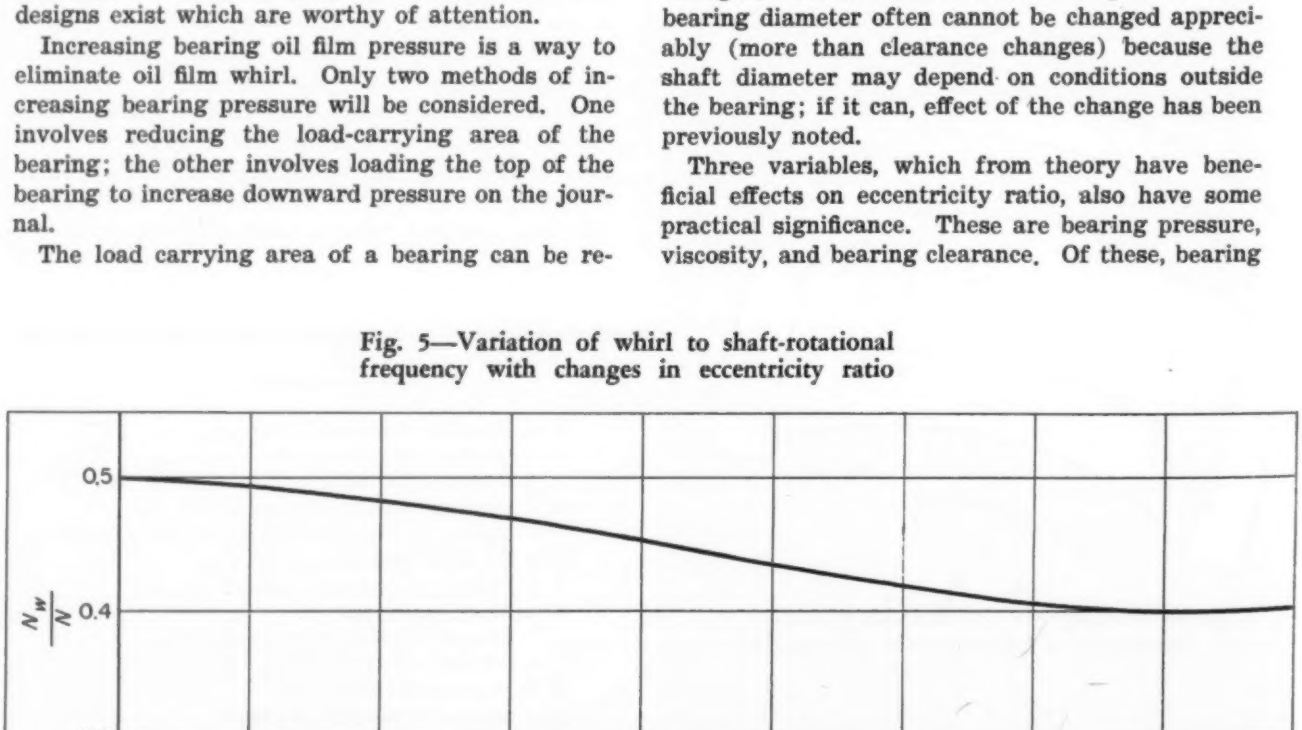
Shortening the length of a bearing is practical and relatively easily done if the original design gave any consideration to this possibility. Reducing the length of a bearing has more effect on increasing the bearing oil film pressure than might be expected at first glance. In bearings where the length-diameter ratio, L/D, is about 1, for example, further reduction of length appreciably increases bearing oil film pressure because the effective load carrying area is abnormally influenced by end leakage.

Newkirk and Grobel[10] developed designs for bearing caps which have the effect of increasing oil pressure in the top half of the bearing. One of these is shown in *Fig.* 4. Such a construction is suitable for rotation in only one direction. Also, it is not suitable for ring-oiled bearings or bearings in which the oil must be admitted in the center of the bearing cap.

In the Appendix, theoretical consideration of remedies for oil film whirl show the beneficial effects of large eccentricity ratios. In that section theoretical consideration is given to variation of bearing pressure, oil viscosity, shaft speed, clearance, and bearing diameter. Since at this point practical considerations are being discussed, some of these theoretical items can be eliminated immediately as being impractical. A design or machine may have to run at constant speed. However, if shaft speed can be changed, a reduction should be contemplated. Also, bearing diameter often cannot be changed appreciably (more than clearance changes) because the shaft diameter may depend on conditions outside the bearing; if it can, effect of the change has been previously noted.

Three variables, which from theory have beneficial effects on eccentricity ratio, also have some practical significance. These are bearing pressure, viscosity, and bearing clearance. Of these, bearing

Fig. 5—Variation of whirl to shaft-rotational frequency with changes in eccentricity ratio

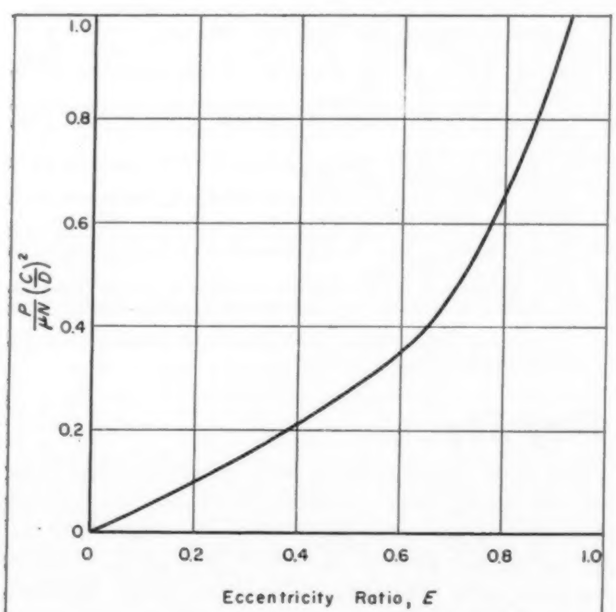


pressure has been covered separately. Sometimes oil viscosity depends on outside conditions such as when the same oil is used to lubricate the gear teeth and the shaft bearings of a speed reducer. On many installations, however, it is perfectly possible to experiment with viscosity changes as a means of eliminating oil film whirl. In fact, experimentation may be the only way to determine accurately the effect of viscosity change on oil film whirl; existing theory and practice don't always agree on the end result of a viscosity change. Theory suggests reducing viscosity if oil film whirl exists, accomplished practically either by putting in a lower viscosity oil or by running the existing oil at a higher temperature. If the oil supply system has a heat exchanger, this may be regulated to permit a higher operating oil temperature. As always, proper judgment must be applied to such a remedy. Reducing viscosity reduces film thickness, which is desirable up to a point. The film must be thick enough to pass foreign matter in the oil safely, for example. Going to the left of the knee of the curve in *Fig.* 7 can be disastrous.

From a practical viewpoint, it would not be too difficult to change the clearance of a bearing which operates with oil film whirl. The question is whether to increase or decrease the clearance for beneficial results. It is perfectly possible to show a desirable effect from increasing clearance on eccentricity ratio (see Appendix). But experimental verification of the end results of a clearance change is lacking, so it is only prudent to proceed carefully. If the first remedial action must be correct, some other method should be tried, such as increasing bearing oil film pressure.

Summary: Several conclusions seem worthy of emphasis. Oil film whirl, although possessing some

Fig. 6—Eccentricity ratio relation for an ideal full bearing



similarity to other bearing disturbances, is characterized by a rotation of the shaft about the center of the bearing at a frequency which is normally less than half the shaft rotational frequency, and is in the same direction as shaft rotation.

An outside exciting force is necessary to start oil film whirl in a journal bearing. However, once established, oil film whirl will sustain itself by action of the oil film alone unless sufficient damping is present in the bearing. When lowest resonant shaft frequency is less than half shaft rotational frequency, oil film whirl may be sustained indefinitely.

Increased bearing pressure produces a larger eccentricity ratio and normally has the effect of stopping oil film whirl. Oil film whirl can also be minimized by use of bearing caps which increase the oil pressure in the cap. At the time of design, the lowest resonant shaft frequency should be established at a value greater than one-half the shaft rotational frequency.

Appendix—Theoretical Remedies

When oil film whirl is encountered in a bearing for the first time, the urgent need for a practical remedy may lead to indiscriminate use of the trial and error method. If several factors are changed at the same time, as is done on occasion, causes and effects are often difficult to set in proper relation. Therefore, it is highly desirable to have sufficient grasp of theoretical considerations to evaluate practical remedies and actions effectively.

Bearing eccentricity ratio is a characteristic which serves as an excellent vehicle for theoretical study of variables influencing oil film whirl. Eccentricity ratio for any given journal and bearing varies exactly as the distance between journal and bearing centers changes. Further, there is a relationship between ratio of oil film whirl frequency to shaft rotational frequency and the eccentricity ratio.

The results of work done by Swift[11] on variation of ratio of oil film whirl frequency N_w to shaft rotational frequency N as the eccentricity ratio changes are shown in *Fig.* 5, for an ideal bearing of infinite length. In an actual bearing, the ratio of N_w/N would be lower at each eccentricity ratio than is shown by the figure; in

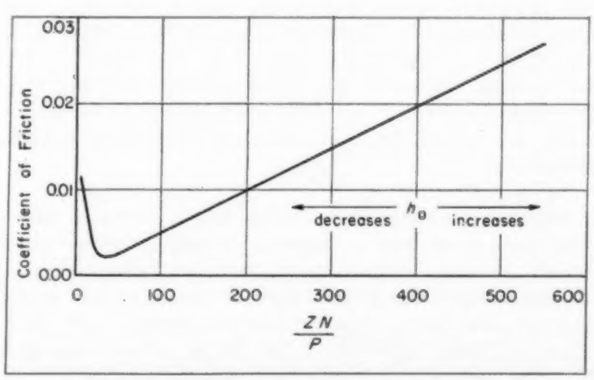


Fig. 7 — Change in coefficient of friction with ZN/P for a full journal bearing

reality the lowest value of N_w/N would be more like 0.3 rather than 0.4.

A most desirable condition prevails when N_w/N is as low as possible, simply because severity of oil film whirl will be minimized if it does occur. To achieve the lowest value, *Fig.* 5 indicates a preference for larger values of eccentricity ratio E.

The question may be asked whether it is possible to change the eccentricity ratio enough to have real beneficial effect on oil film whirl frequency. Assume a reasonable possibility of increasing E from 0.4 to 0.6 in *Fig.* 5. At $E=0.4$, $N_w/N=0.45$; at $E=0.6$, $N_w/N=0.420$. If the shaft has the same rotational frequency in either case, the higher eccentricity ratio produces an oil film whirl frequency about 7 per cent lower than at the smaller eccentricity ratio. This is worth going after, because frequently the speed at which a shaft may whirl is a matter of 10 per cent higher than the speed at which whirl disappears.

At this point, the variables that have an effect on eccentricity ratio E must be considered. Eccentricity ratio is a function of bearing pressure P, viscosity μ, speed N, clearance C, and journal diameter D. As a dimensional equality (not numerical), this relationship can be expressed as

$$E = \phi\left[\frac{P}{\mu N}\left(\frac{C}{D}\right)^2\right] \qquad (1)$$

where ϕ indicates function. The arrangement of the several variables in Equation 1, resulting from the technique of dimensional analysis, is clearly presented by Faires.[12] *Fig.* 6 taken from work by Howarth[13] indicates the numerical relationship between E and the dimensionless group for an ideal full bearing.

Preference is for larger values of eccentricity ratio E, as outlined previously; results from changes in P, μ, N, C and D can be evaluated from *Fig.* 6. At the same time, *Fig.* 7 is of interest because it enables us to take friction loss and minimum film thickness into account. *Fig.* 7 is a slight variation of a table by McKee.[14] Also, eccentricity ratio can be expressed as:

$$E = 1 - 2\frac{h_0}{C} \qquad (2)$$

This equation results from straight forward geometry of position of the journal with respect to the bearing.

Bearing Pressure Variation: Inspection of the dimensionless symbol group in Equation 1 indicates immediately that increasing P is a step in the right direction, as it gives not only the preferred higher values of E (see *Fig.* 6) but the lower and better values of N_w/N. Simultaneous consideration of *Fig.* 7 shows that as P increases (Z and N constant) friction loss decreases which is desirable, and h_0 also decreases. Also, inspection of Equation 2 shows decreasing h_0 to result in a larger eccentricity ratio E if clearance C is constant. Thus this reasoning simply confirms what is already known; if oil film whirl occurs in an actual situation, it can be minimized by increasing bearing pressure.

Oil Viscosity Variation: Consideration of *Fig.* 6 shows that as μ decreases a larger eccentricity ratio results which, in turn, leads to more-desirable lower values of N_w/N. And *Fig.* 7 indicates a satisfactory action through decrease of friction loss and thinner film thickness h_0. From Equation 2 the effect of decreasing h_0 is to increase eccentricity ratio which is helpful in relieving oil film whirl. Increasing oil temperature, thus decreasing viscosity, might also be a help in reducing oil film whirl under certain circumstances.

Shaft Speed Variation: There is no need to belabor this variable, since the line of reasoning is becoming familiar. Suffice it to say that, if possible, journal speed N should be decreased when oil film whirl is encountered. Decreasing N also leads to larger eccentricity ratio and preferred lower N_w/N.

Clearance Variation: A look at *Fig.* 6 suggests that increasing clearance might be helpful as a means of getting a more favorable N_w/N by increasing eccentricity ratio. However, Equation 2 shows that what happens to h_0 and bearing stability when both E and C are changed is not so clear, particularly when the rate at which C and E changes are different. Although several numerical examples which the writer has worked out yield smaller h_0 for an increase in clearance C it appears wise to compute the effect on h_0 when considering an increase of clearance. By a different approach, Hagg[15] comes to a similar conclusion regarding the effect of changing clearance. In general, increasing values of C/D are attended by a decrease in friction loss and film thickness; this holds if Z, N and P are steady.

Bearing Diameter Variation: Effects of a change in bearing diameter D can be explored in several ways. If other factors are constant, decreasing diameter increases bearing pressure, which may be considered desirable. In considering the effect of changes in clearance, the general effect of C/D changes has already been noted. Therefore, it would appear that decreasing diameter would have beneficial effects on friction loss as well as resulting in a smaller h_0.

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They Say . . .

"In your preoccupation with your slide rule you cannot safely leave to others—to the economists and politicians and lobbyists—the larger job of defending the way of life which was made possible by your own contribution."—Charles E. Wilson, *General Electric Co.*

hard vinyl plastics

A structural material with good corrosion resistance, nonplasticized polyvinyl chloride is easily fabricated by welding or machining

By M. J. Pagerie

Vice President
American Lucoflex Inc.
New York

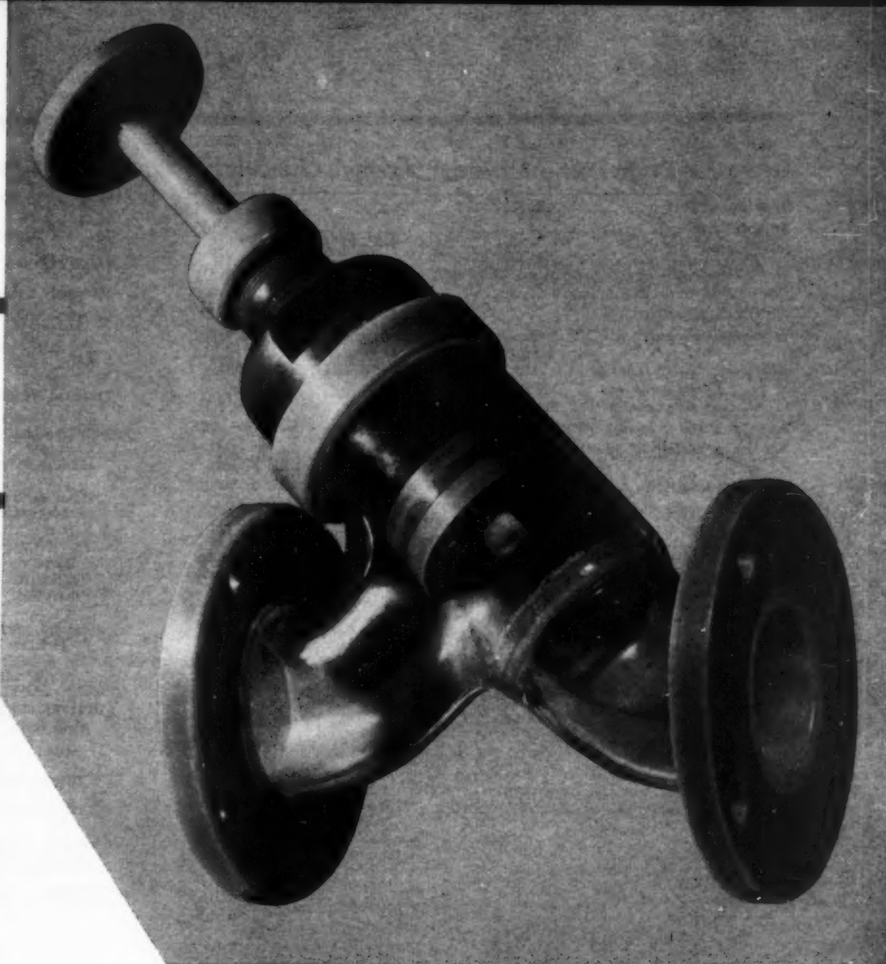


Fig. 1—Above—Valve for heavy chemical operations, fabricated entirely from nonplasticized polyvinyl chloride

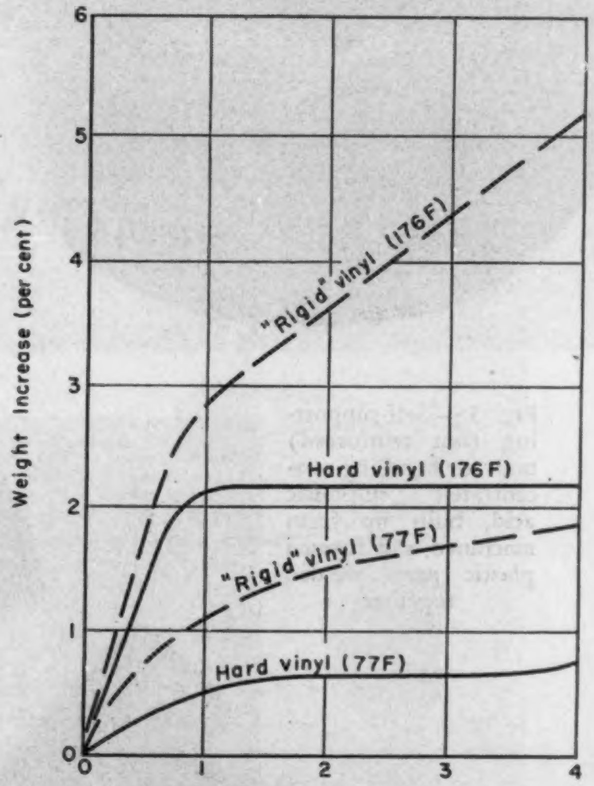


Fig. 2—Weight increase after immersion in water for hard vinyls and plasticized "rigid" vinyls

FINDING wide use today as corrosion-resistant structural materials, *Fig.* 1, nonplasticized vinyls have several outstanding properties, among them excellent stability in the presence of moisture, easy machinability and weldability, and resistance to a broad range of chemicals. Hardness and structural rigidity in hard polyvinyl chloride are due principally to the omission of plasticizers which can bleed out, causing a spongy structure. Thus, in many of their properties, nonplasticized hard vinyls surpass the so-called "rigid" vinyls, which do contain plasticizers.

Low water absorption of nonplasticized vinyls, compared with typical rigid vinyls, is shown for two different temperatures by *Fig.* 2. The higher temperature of 176 F is the recommended maximum operating temperature for the nonplasticized vinyls. A thermoplastic, hard polyvinyl chloride has good structural properties, TABLE 1, which make possible the use of rigid assemblies without structural support other than that supplied by the

Table 1—Properties of Hard Polyvinyl Chloride

Specific gravity	1.4
Tensile strength (psi)	8600
Elongation (%)	6.2
Modulus of elasticity (psi)	3.5×10^{5}
Compressive strength (psi)	11,000
Flexural strength (psi)	13,200
Izod impact strength, flatwise (lb-in./in. notch)	0.761
edgewise (lb-in./in. notch)	0.571
Hardness, Rockwell R	121
Scleroscope	65
Durometer D	86
Heat distortion point (F)	177
Linear coefficient of expansion (per deg C)	6×10^{-5}
(per deg F)	$3.3\text{-}4.5 \times 10^{-5}$
Specific heat	0.32
Thermal conductivity (Btu/hr/ft²/deg F/in.)	2.8×10^{-4}
Char temperature (F)	430

Fig. 3 — Self-supporting (not reinforced) tank for handling concentrated sulphuric acid, built up from machined and formed plastic parts welded together

Table 2—Chemical Resistance of Nonplasticized Vinyls*

Reagent†	— Temperature —		
	20 C	40 C	60 C
Mineral bases			
Ammonia (100)	X	X	X
Ammonium hydroxide (dil.)	X	X	
Potash (40)	X	X	
Soda (40)	X	X	
Mineral acids			
Arsenic anhydride (dil.) (80)	X	X	
Carbonic (all)	X	X	X
Chromic (all)	X	X	
Hydrochloric, gas (all)	X	X	X
aqueous (30) (37)	X	X	X
Hydrofluoric (46)	X		
Hydrofluosilicic (30)	X		
Nitric (all)	X	X	
2/3 Nitric (70) plus 1/3 hydrofluoric (46)	X		
Perchloric (72)	X		
Phosphoric (30)	X	X	X
(85)			X
Selenic (sp gr 1.4)	X		
Sulphuric (40-80)	X	X	X
(95)			X
Sulphur dioxide, gas or sol. (all)	X	X	
Mineral salts			
Aluminum chloride (dil.)	X	X	
(sat.)	X	X	X
sulphate (dil.)	X	X	
Ammonium chloride (dil.)	X	X	
nitrate (sat.)	X	X	X
sulphate (dil.)	X	X	
(sat.)	X	X	X
Borax (sat.)			X
Copper sulphate (dil.)	X	X	
(sat.)	X	X	X
Ferric chloride (dil.)		X	
Potassium bichromate (sat.)	X		
chloride (sat.)	X	X	X
cyanide (sat.)	X	X	X
Zinc chloride (dil. or sat.)	X	X	X
sulphate (dil.)	X	X	
Organic products			
Acetic acid, aqueous (25-85)	X	X	
(95)	X		
(99.5)	X		X
Butyl acetate (100)	X		
alcohol (100)	X		
Carbon tetrachloride (100)	X		
Chloroacetic acid (100)	X	X	
Citric acid (10)	X		
Ethyl alcohol (dil.) (100)	X	X	
Fatty acids (100)	X		
Formaldehyde (all)	X	X	X
Formic acid (50) (100)	X	X	
Glycerine (all)	X	X	X
Lactic acid (100)	X		
Lauryl chloride (100)	X		
Methyl alcohol (100)	X	X	
Oxalic acid (dil.)	X	X	
Miscellaneous			
Hydrogen peroxide (30)	X		
Photographic developers (various)	X	X	
Sea water (100)	X	X	

*X indicates resistant. †Concentration in parentheses.

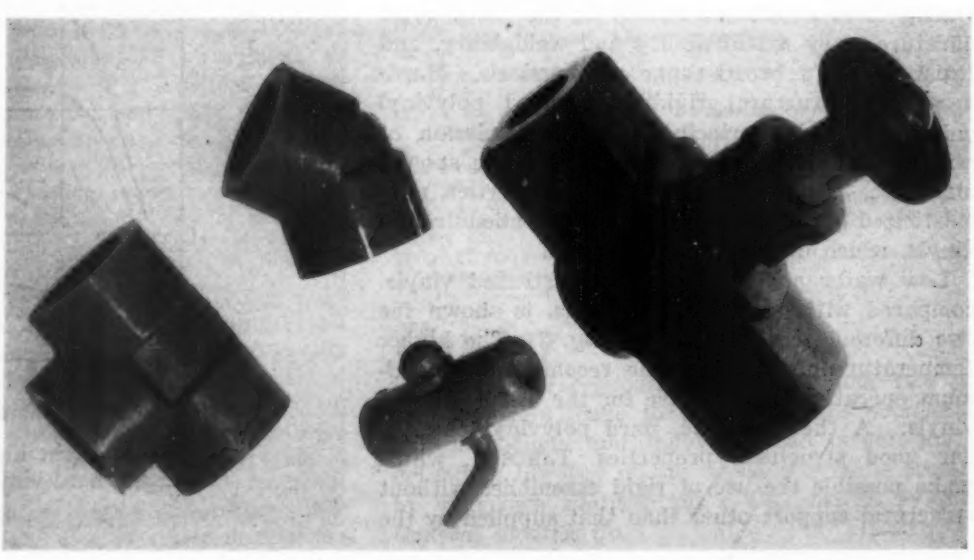


Fig. 4—Typical hard vinyl valves and fittings

Table 3—Electrical Characteristics of Hard Vinyls

Dielectric strength, 60 cycles, short-time, ⅛-in.	419
step by step, ⅛-in.	357
Dielectric constant, 1000 cycles	3.2
10^6 cycles	3.0
Dissipation factor, 1000 cycles	0.02
10^6 cycles	0.01
Loss factor, 1000 cycles	0.07
10^6 cycles	0.04
Arc resistance (seconds)	65

plastic itself, *Fig.* 3. Structural strength is maintained to an adequate degree at elevated temperatures; tensile strength is 13,800 psi at −55 F, and 2900 psi at 160 F.

At ordinary temperatures, hard polyvinyl chloride is unaffected by mineral acids, bases, salts, chlorides, oils, grease, gasoline and carbon tetrachloride in various concentrations. It is, however, susceptible to some organic solvents, such as aromatic and chlorinated hydrocarbons, esters, ethers and particularly ketones. Resistance to a number of common chemicals is shown in TABLE 2. Because of its resistance to aging and oxidation it is suitable for many applications where oxidative corrosion is a problem.

The plastic is self-extinguishing and nontoxic. Translucent in thin sections and slightly opaque in thick sections, hard vinyls have an amber to light or medium-tan color. Electrical characteristics are shown in TABLE 3.

Forms: Nonplasticized polyvinyl chloride is fabricated primarily by welding or machining. Granular resins are, however, available for molding and extrusion. Typical forms and maximum size ranges are:

Sheet—1/64 to 1 inch thick
Block—3 inches thick, 20 by 20 inches
Bar—to 2½ inches diameter
Tubing—OD to 6 inches
Welding rod—triangular, 0.20 and 0.28-inch sides

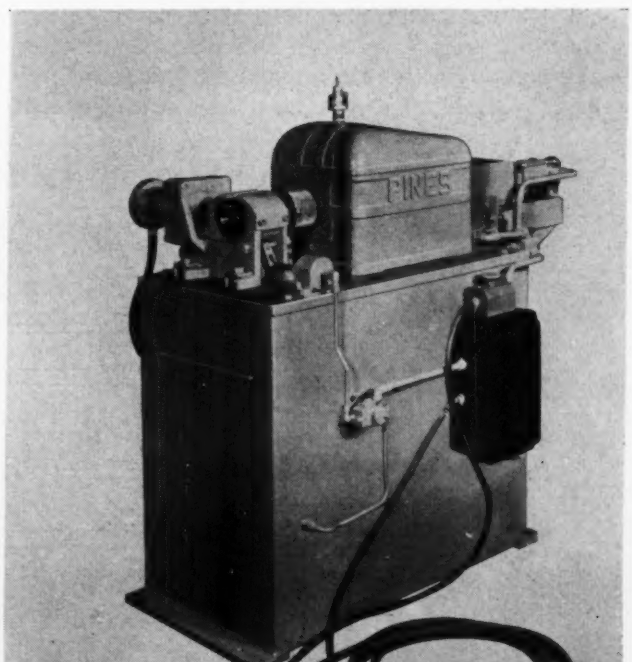
Additionally, fittings and valves for piping applications requiring corrosion resistance, *Fig.* 4, are also available.

Welding: Average standard sheet thickness for welded assemblies such as liners and inside ducting is ⅛-inch. Sheets are joined by using rods of 0.060 to 0.160-inch thick material or of standard triangular shape with the aid of a hot-air jet gun, which directs a restricted flow of preheated air at the weld point. Welds of this type, if properly done, have been found to be better than 80 per cent efficient. The weld, which is completed in one pass, is homogeneous, since rod and material are identical; there is no brittle area or zone of chemical or physical weakness as with metals.

Machining: Sawing, drilling, cutting, shaping, planing and milling are accomplished on conventional equipment. Hard vinyls, however, are thermoplastics, and surface softening occurs if heat is developed too rapidly during any machining operation. Rate of heat development is dependent primarily on tool design, sharpness, cutting speed and feed. Turning, facing, boring and chasing operations can be carried out satisfactorily if ordinary metal cutting tools are modified by increasing front and side clearances to reduce frictional heat and facilitate chip removal.

Pneumatic Operation Speeds Finishing

PNEUMATIC chuck closing, spindle advance and return combine to reduce cycle time, not including cutting time, of a new end-finishing machine, to 1.5 seconds. Chuck and spindle air cylinders are controlled by a foot switch. Movement of the switch closes the chuck, the work locating stop is pivoted clear by a small air cylinder mounted on the side of the clamping unit and the spindle is advanced through the machining cycle. An adjustable stop automatically retracts the spindle when the cut is completed.

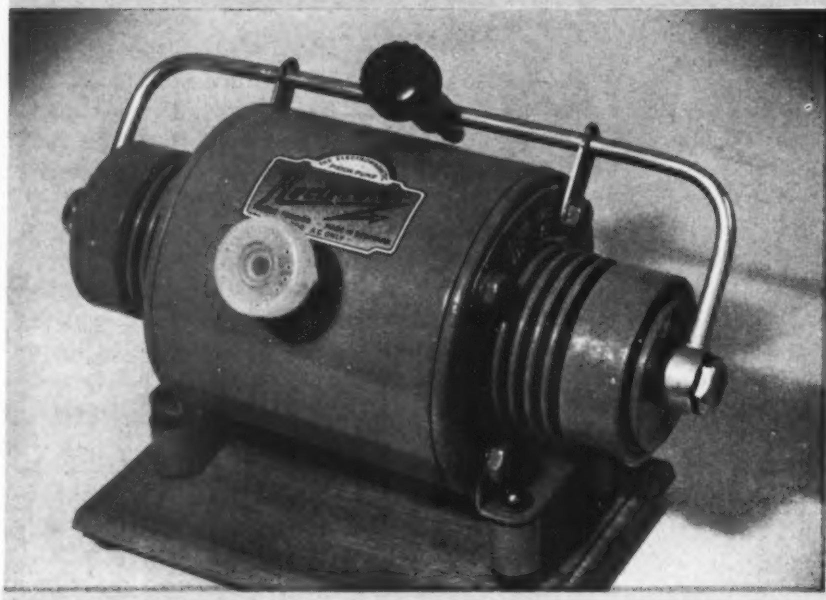


Designed for work up to 3 inch OD, the Pines Series 700 machine has a 2-inch stroke and maximum spindle speed of 2000 rpm. Applications include light turning chamfering and pointing bar stock, shaping tube or rod ends and center drilling.

Total space required for the machine is 26 inches wide, 57 inches long and 45⅝ inches high. Machine dimensions are 20 inches wide and 39 inches long with 6-inch chuck overhang on the width and 18-inch overhang of the spindle unit to the rear. Weight, complete with pneumatic cylinder, is 850 pounds.

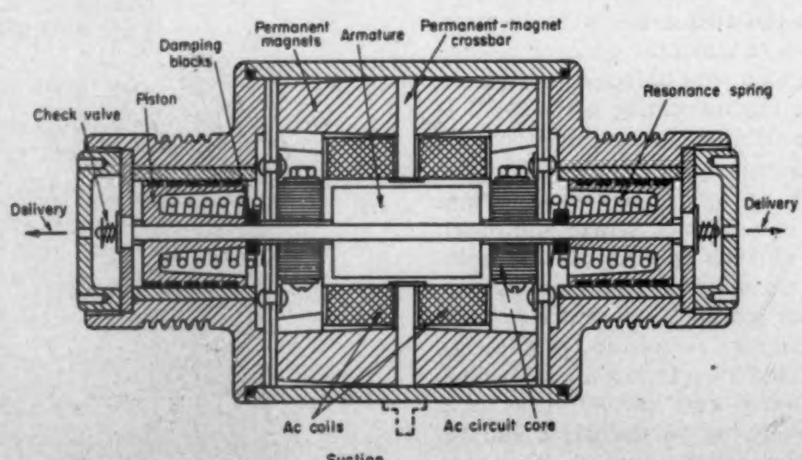
CONTEMPORARY DESIGN

Vibrating Pistons Supply Air Pump Pressure



A RECIPROCATING armature with a piston on each end is the pumping element of Reciprotor air pumps supplied by Gram & Anderson I/S of Copenhagen, Denmark. Operating from 220 volts ac, the pumps have a power consumption of about 60 watts, although this consumption can be reduced by use of a sliding rheostat or variable transformer to about 5 watts. Pumps are available with piston diameters of $1\frac{3}{4}$, $1\frac{1}{2}$ or $1\frac{1}{4}$ inches, and different models have been designed for continuous (8-hour), normal (1 to 2 hours of continuous use), and intermittent ($\frac{1}{2}$-hour) service. Maximum pumping capacity is 1.4 cubic feet per minute, depending upon type of pump; maximum pressure is 30 inches of mercury.

Vibrator consists of an armature which is magnetized by a system of permanent magnets, thus eliminating the need for rectification of the alternating current to separate it into positive and negative pulses. Oscillating at the frequency of the ac power source, the armature and pistons have a maximum stroke of $\frac{1}{4}$-inch. Heavy springs provide the necessary resonance and permit greater vibration at normal pumping loads. Check valves are provided in each piston (not shown) and at the ends of the cylinders. Nylon piston rings require no lubrication. Cooling is accomplished by the air passing through the pump. Rubber damping blocks and rubber mounting cushions help minimize vibration. Although cylinder sleeves are hard chrome-plated steel and permanent magnets are special steel, most working parts are of light metal, protected by cadmium plating, or light-gray varnish on exterior parts.



Transactions of

The First Conference on

MECHANISMS

Purdue University, West Lafayette, Ind.
October 12 and 13, 1953

Providing a meeting ground for all those concerned with

- **the design of mechanisms**

this first Conference was planned

- **to sharpen focus on essential elements in design,**
- **to present realistic design methods,**
- **to describe uniquely successful applications.**

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Transactions of

The First Conference on

Mechanisms and Their Classification

By Allen S. Hall Jr.
*Professor of Mechanical Engineering
Purdue University
Lafayette, Ind.*

THE past ten or fifteen years have seen in this country an accelerating increase in attention to the improved design of mechanisms. This has been stimulated in part by need for improved performance as operating speeds were increased, with attendant inertia problems, and in part by the rapidly growing importance of automatic computing and control equipment. The increasing use of fluid and electrical equipment, instead of tending to render mechanical devices obsolete, seems to have created new demands for them. In some cases new respect for the simplicity and reliability of well-designed mechanical systems has been generated by the expensive trial substitutions of more "modern" electronic equipment.

Although the art of devising mechanisms is old, it is by no means old-fashioned. The field is far from sterile. You who are designing better cams are as much leaders in technical progress as the man who designs a better television circuit.

The beginnings of the machine designer's art

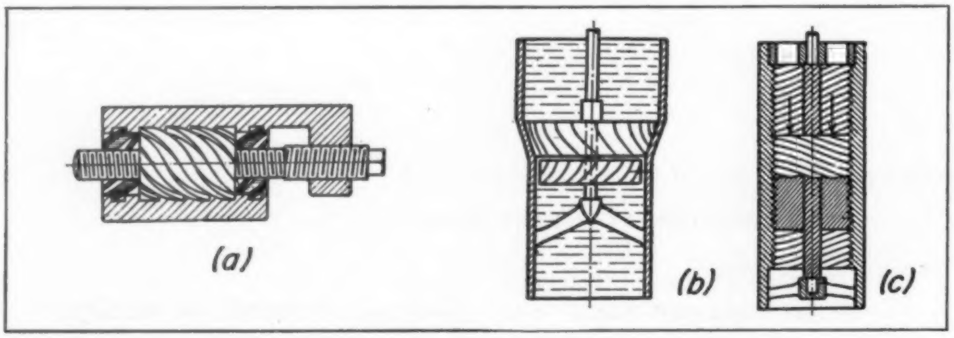
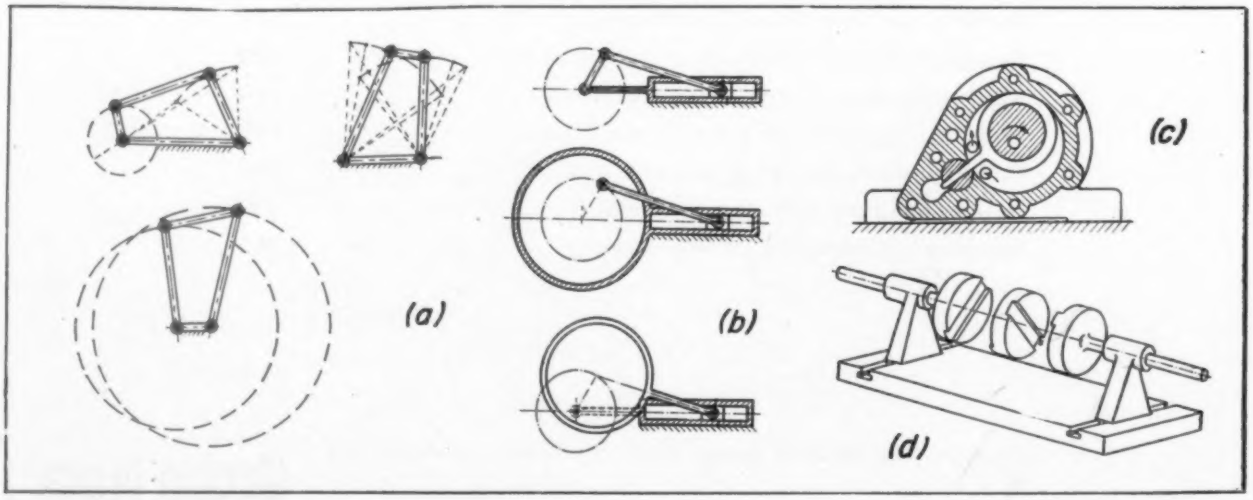


Fig. 1 — Screw mechanisms: *a*, general screw mechanism; *b*, a fluid screw mechanism; *c*, its rigid body equivalent

Fig. 2—Below—Plane crank mechanisms of which the simplest, and at the same time the most important, are the 4-bar linkage and the slider-crank: *a*, inversions of the 4-bar linkage; *b*, variations of the slider-crank through expansion of pins; *c*, a pump mechanism consisting of an inversion of the slider-crank with expanded pins; *d*, Oldham coupling, one form of the double slider mechanism



MECHANISMS

are lost in prerecorded history. The use of rollers for moving heavy objects, the bow and fire drill, and the wheel are probably among the earlier accomplishments. The art progressed to impressive heights with little assistance from the basic sciences. We know, for example, that quite satisfactory toothed gearing was being built long before the theory of tooth shapes was explained mathematically.

Reuleaux: "Mechanisms" or "applied kinematics", as a distinct area of study with some reason to be called a science, is generally considered to have started around 1875, with the publication by Franz Reuleaux of the first volume of his *Theoretische Kinematik*.[1] This was translated into English the following year by A. B. W. Kennedy[2] and has had a great influence on all subsequent texts.

Prior to Reuleaux machinery seems to have been studied by description of complete machines. Little attempt was made to analyze basic elements common to different machines. Each machine was treated as something unique. Of course, to such study there is no logical beginning or ending, there is much repetition, and little learning of anything that could be called "fundamentals" of the subject.

Reuleaux put the study of machinery on a logical foundation by (1) defining "machine" and "mechanism" in a meaningful, unambiguous manner, (2) pointing out basic building blocks from which all mechanisms are formed, and (3) establishing a classification of known mechanism types. Reuleaux's contributions went beyond these, but these are what we wish to examine briefly in this discussion.

Reuleaux defined a machine as "a combination of resistant bodies so arranged that by their means the mechanical forces of nature can be compelled to do work accompanied by certain determinate motions"

[1]References are tabulated on Page 180.

Fig. 3—Space crank mechanisms: *a*, spherical 4-bar linkage which in one form is the familiar universal joint, or Hooke's coupling; *b*, spherical equivalent of the plane slider crank; *c*, a "wobble" mechanism; *d*, a general 4-link space mechanism containing two pin joints, one cylinder joint and one ball joint

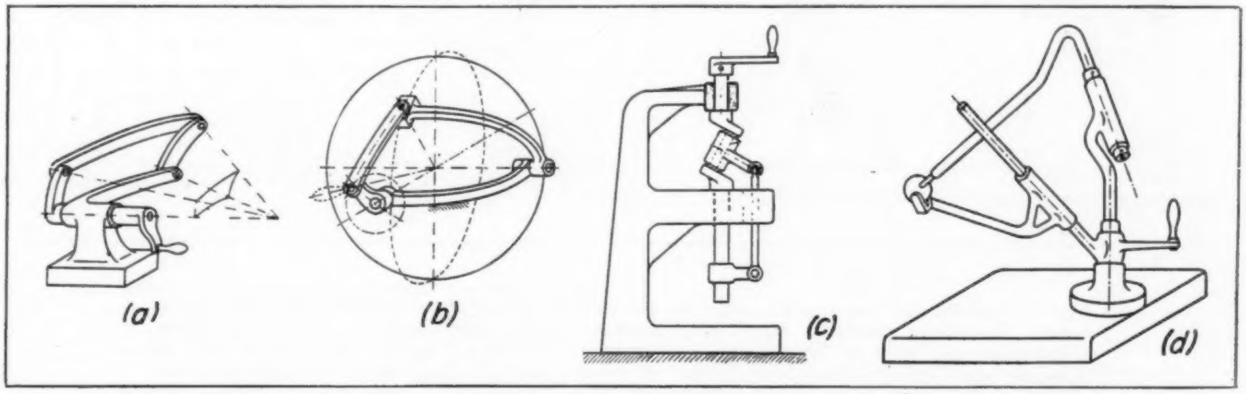
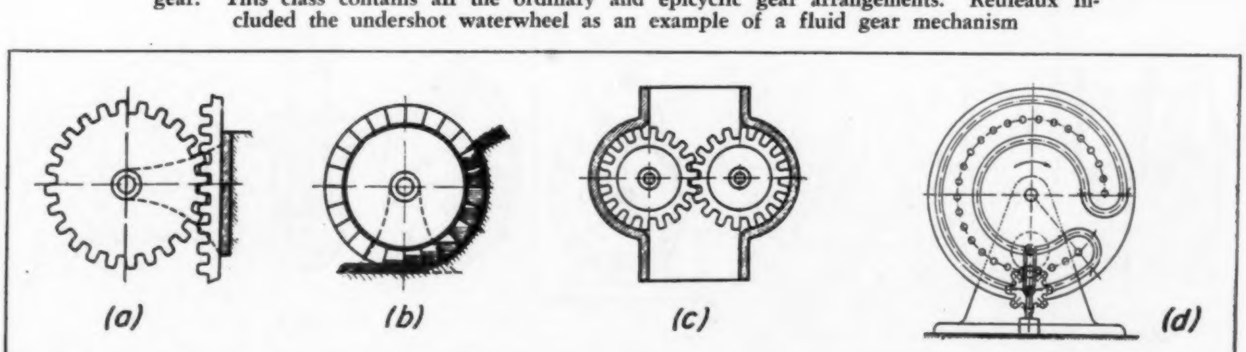


Fig. 4—Gear mechanisms: *a*, pinion and rack; *b*, "fluid gear"; *c*, gear pump; *d*, mangle gear. This class contains all the ordinary and epicyclic gear arrangements. Reuleaux included the undershot waterwheel as an example of a fluid gear mechanism



(as translated by Kennedy).

The textbooks we use today stick quite closely to the Reuleaux definition. For example, according to Ham and Crane[3], "A machine is a combination of rigid or resistant bodies, so formed and connected that they move upon each other with definite relative motions and transmit force from the source of power to the resistance to be overcome." A mechanism . . . "a combination of rigid or resistant bodies, so formed and connected that they move upon each other with definite relative motions".

The term "mechanism" is used when we fasten attention on the motion problem, sufficient strength and rigidity being assumed. A machine may, and usually does, consist of a number of mechanisms in series and/or parallel combinations, working together in synchronization.

Reuleaux recognized the pre-eminence of the *joints* between bodies as the basic elements in the composition of mechanisms. He called attention to the "pairs" of geometric elements in which two resistant bodies make contact, and distinguished between "higher pairing" (point or line contact) and "lower pairing" (surface contact). He also recognized and placed great emphasis on the fact that many apparently different mechanisms are basically identical (identical in the basic chain of resistant bodies), differing only through expansion of pairs (example, crank versus eccentric), inversion of lower pairs (exchange of hollow and solid elements), and inversion of the mechanism (fixing a different link of the basic chain). He mentions 30 different patents for "rotary" engines, all based on the slider-crank mechanism.

Reuleaux placed the known basic types of mechanisms in six classes:

1. Screw mechanisms (Schraubentrieb)
2. Crank mechanisms (Kurbeltrieb)
3. Gear mechanisms (Radertrieb)
4. Pulley mechanisms (Rollentrieb)
5. Cam mechanisms (Kurventrieb)
6. Locking mechanisms (Gesperrtrieb)

Accompanying illustration, *Figs.* 1 to 7 from Beyer[4] show some typical members of each class.

Some such classification scheme is essential as a basis for organized study of mechanisms. It would obviously be impossible to even read descriptions of more than a small percentage of the machines that have been invented. But with an adequate classification system as a framework, it is possible to organize a study to include typical mechanisms of each class, thus insuring a reasonably thorough coverage of the subject. Many of our mechanism textbooks use essentially the Reuleaux classes as a set of chapter headings.

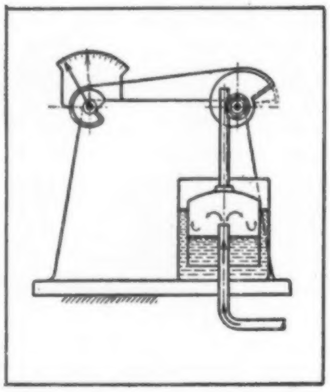


Fig. 5 — Noncircular pulleys used in a gas velocity meter. The class of pulley mechanisms contains all the belt, rope and chain drives, conveyors, hoisting devices, etc.

Fig. 6—Below—Cam mechanisms

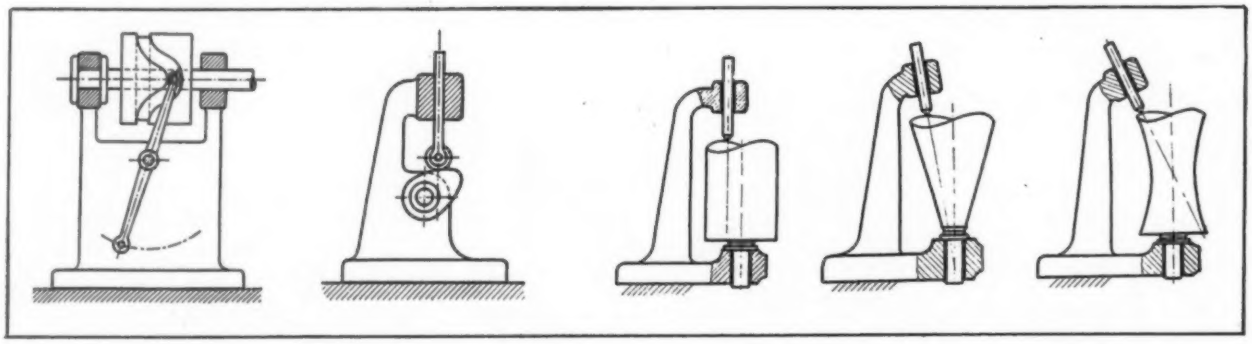
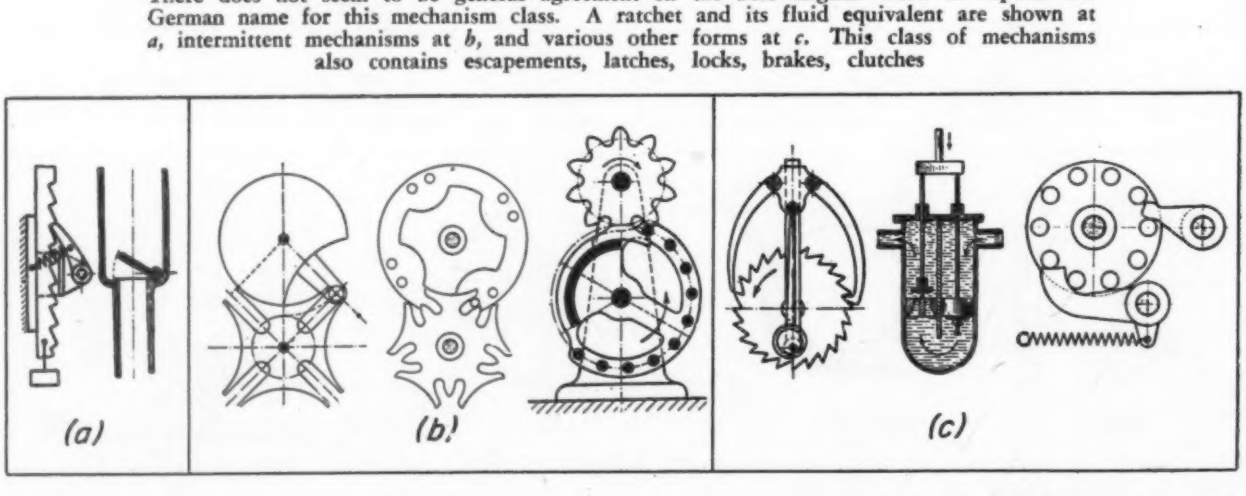


Fig. 7—Locking mechanisms, so called by the present author because these "locking" mechanisms all contain provision for locking against motion under some set of circumstances. There does not seem to be general agreement on the best English word to replace the German name for this mechanism class. A ratchet and its fluid equivalent are shown at *a*, intermittent mechanisms at *b*, and various other forms at *c*. This class of mechanisms also contains escapements, latches, locks, brakes, clutches



It is, of course, to be recognized that there are a multitude of mechanisms that do not fit precisely into any one of the Reuleaux classes. One mechanism may contain features typical of two or more classes,

It should also be emphasized that many apparently complex mechanisms turn out, on closer examination, to be composed of two or more separate mechanisms, each of which is a relatively simple example of one of the basic types. It seems to me especially important that anyone (teacher, writer, editor, etc.) attempting to describe machinery for someone else keep this always in mind. Many descriptions of new machines appearing in print are difficult to follow because neither the words nor the drawings are designed to show clearly the separate mechanisms involved, *Fig.* 9.

Many subclasses in the Reuleaux scheme have been investigated in detail by later writers. The aim of such investigations has been to display all conceivable members of the subclass. An example is the paper on "release gears" by De Jonge[5]. *Figs.* 10 and 11 are taken from this source. Investigations of this kind constitute one form of "kinematic synthesis".

Franke: In 1942 Franke[6] made suggestions for major revisions or extensions in the concept of a mechanism. His suggestions have not all won universal acceptance, and may never do so, but are nevertheless stimulating as a fresh viewpoint.

In his book *Vom Aufbau der Getriebe* he proposed these definitions: "A mechanism is a device for the coupling and transformation of energies". "A machine is a mechanism with at least one mechanically moved part."

The word "mechanism" is here a loose translation of the German "Getriebe". These definitions include a number of things not included in Reuleaux's. Examples are (1) electrical systems, (2) fluid dynamic systems, (3) mechanical systems in which the relative motions of the parts are not completely determined by the forms, arrangement, and connections of the parts, but are dependent in addition on inertia, friction, gravity and spring forces. Franke's classification of mechanisms is illustrated by *Fig.* 12 taken from his book.

His main classes are based on constraint: (1) constrained, one degree of freedom in the relative motions, (2) partially constrained, devices in which relative motions are partially dependent on inertia,

Fig. 8—The arrangement of links and gears in the "Pilgrim Step" mechanism, which would not fit exactly any one of the Reuleaux classes. It contains features typical of both the crank mechanisms and the gear mechanisms

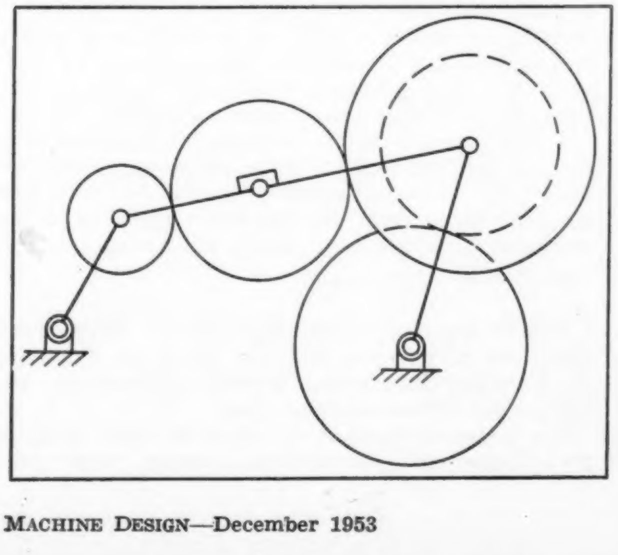


Fig. 9—Cam-operated sliding block mechanism shown at *a* can be described more easily as at *b* where a skeleton drawing is shown and the three mechanisms, which in series combination form the total mechanism, are shown

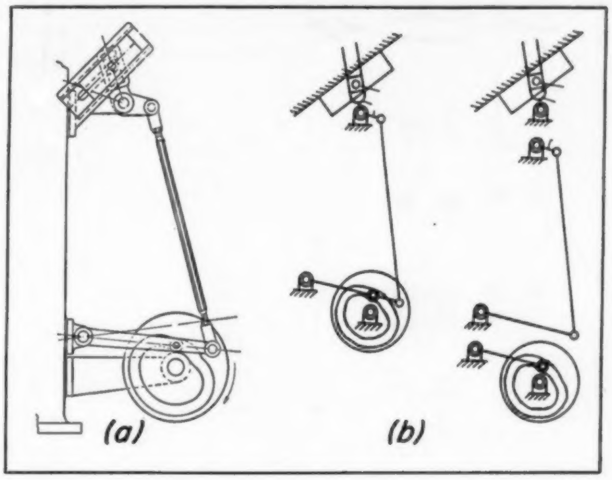


Fig. 10—Six subclasses of Reuleaux's sixth class as displayed by De Jonge

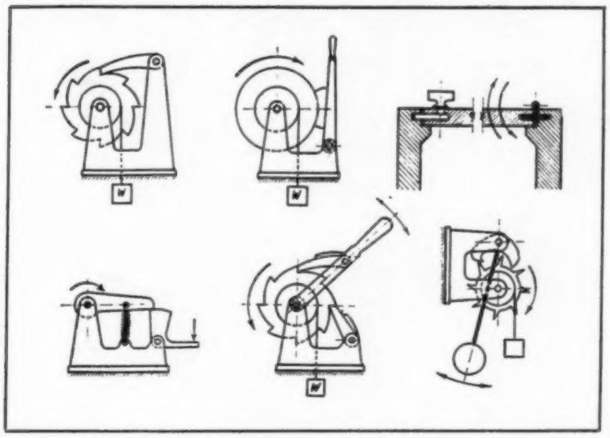
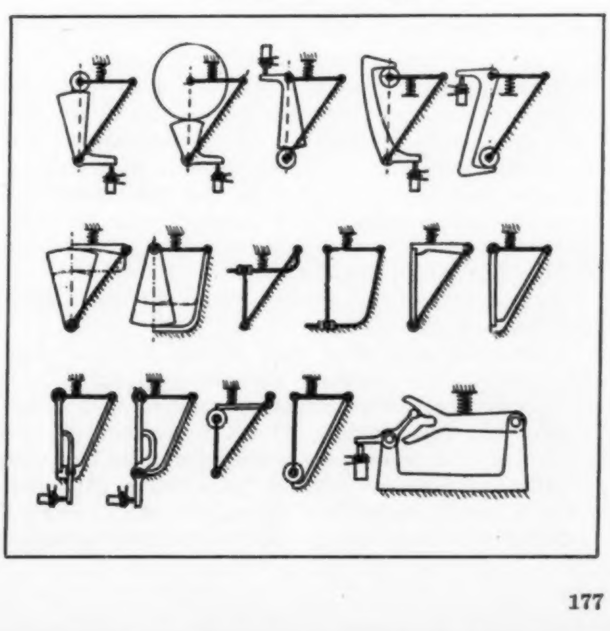


Fig. 11—Some of the possible forms of "release gears" displayed by De Jonge



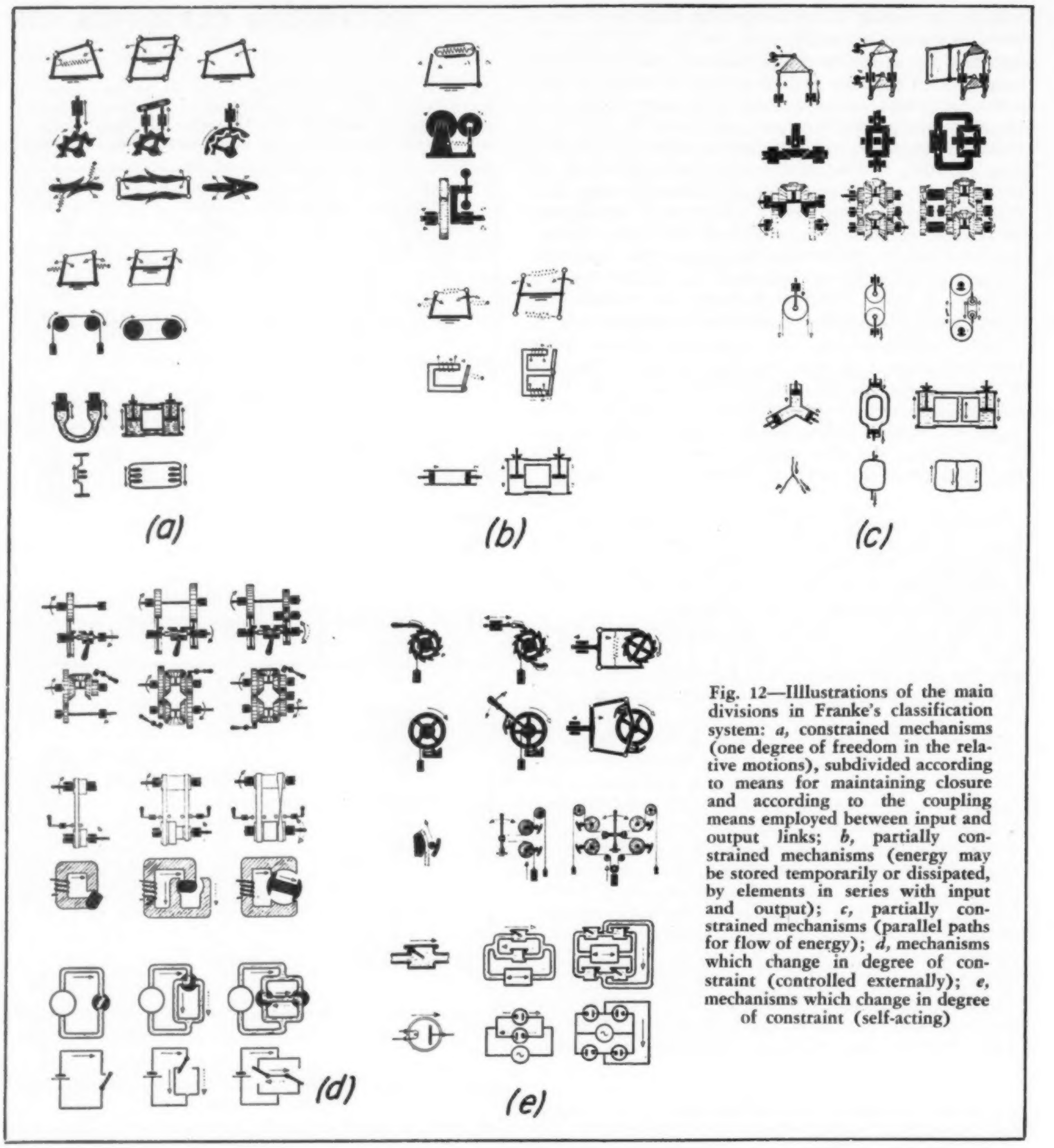


Fig. 12—Illustrations of the main divisions in Franke's classification system: *a*, constrained mechanisms (one degree of freedom in the relative motions), subdivided according to means for maintaining closure and according to the coupling means employed between input and output links; *b*, partially constrained mechanisms (energy may be stored temporarily or dissipated, by elements in series with input and output); *c*, partially constrained mechanisms (parallel paths for flow of energy); *d*, mechanisms which change in degree of constraint (controlled externally); *e*, mechanisms which change in degree of constraint (self-acting)

friction, spring forces, etc., (3) devices in which the degree of constraint changes during operation. His first class contains most of the mechanisms covered by Reuleaux's definition, except locking mechanisms.

Franke took the pin joint to be the fundamental element in the composition of mechanisms. In other words he took the generalized form of any mechanism to be a pin-jointed combination, *Figs.* 13 and 14.

His concepts in this respect have been subjected to criticism on two bases:

1. The pin joint is a very special form of joint, inappropriate in studying general combinations. A pin-jointed mechanism cannot be equivalent to a cam mechanism, for example, except in constraint. The equivalence in motion transmission can be only very roughly approximate over a finite range of motion. (For a motion instant the equivalence can be exact in velocity and acceleration.) However, a cam can always be the equivalent of a pin-jointed mechanism. It is possible to design a cam to transmit any motion transmitted by a linkage, but the reverse is not true. Hence a cam mechanism would seem to be a more "general" form than a pin-jointed mechanism.

2. This concept of the pin-jointed arrangement as the general form leads to some unnecessary complications. For example, according to Franke's concepts, the generalized form for the roller guide (a 4-link mechanism), would be a 12-link pin-jointed combination, *Fig.* 15.

Sieker: About six years ago, Sieker[7] voiced some objections to previous schemes (including Franke's) for showing relationships between mechanisms, and made some alternative suggestions.

For plane mechanisms he takes the "roll" joint, or its equivalent in constraint, the two-point "skid" joint,

Fig. 16*a*, to be the basic element. The "slip" joint, *Figs.* 16 *b* and *c*, he shows to be a special case of the skid joint. A further specialization gives the pin joint, *Fig.* 16 *d*.

For him the generalized form of any mechanism is a combination of bodies in which all joints are either roll or skid joints. Thus his generalized three-link plane mechanism is a combination of three rigid bodies having plane relative motion and constrained by two "roll" joints plus one "skid" joint, *Fig.* 17. This combination he considers to be the common ancestor of all three-link plane mechanisms.

In the same spirit he takes the generalized four-link mechanism to be a combination of four bodies and four roll joints. Particular four-link mechanisms are obtained by substituting special joints for the general roll joint, *Fig.* 18.

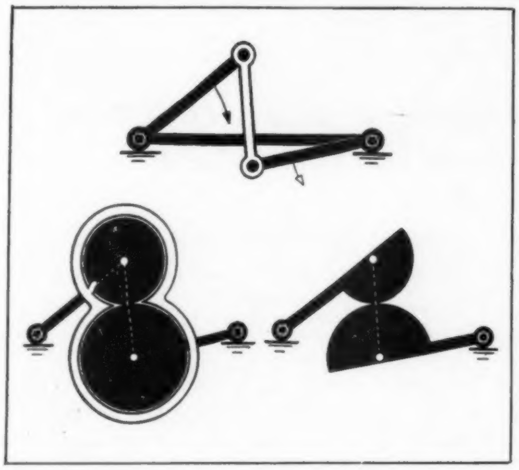


Fig. 13—Derivation of cam-type mechanism from pin-jointed linkage. Part of Franke's argument for considering the generalized form of any mechanism to be a pin-jointed combination of rigid bodies

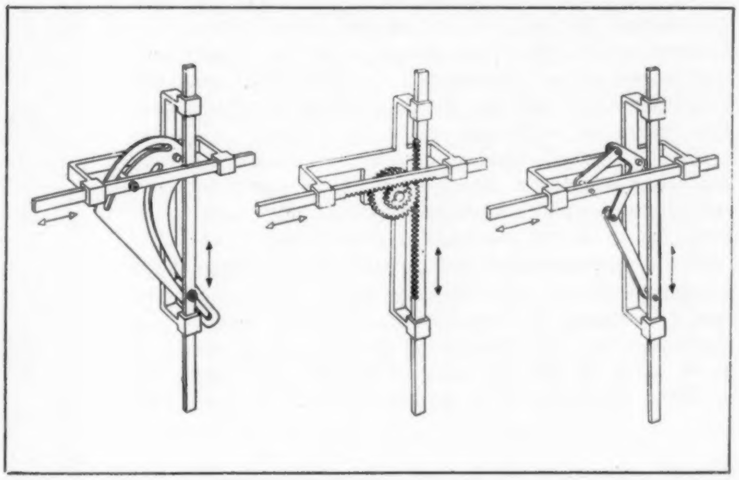


Fig. 14—Three different mechanisms for converting vertical reciprocation to horizontal reciprocation, all having the same 6-link pin-jointed arrangement as the basic chain form, according to Franke

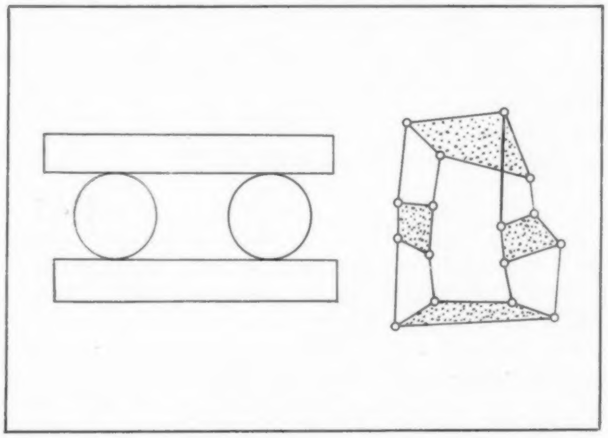


Fig. 15—Roller guide and corresponding chain form in accordance with Franke's point of view

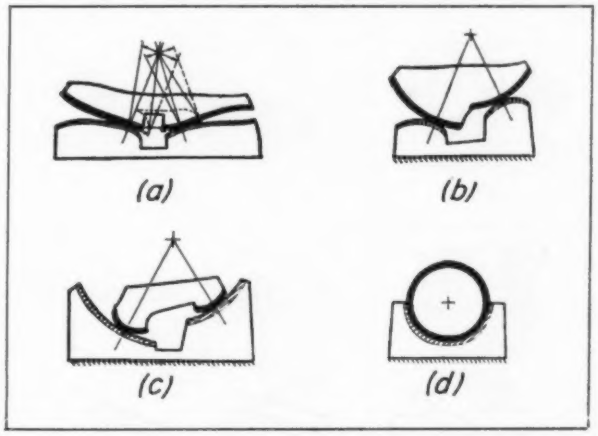


Fig. 16—Evolution of pin joint as specialized form of two-point skid joint, according to Sieker

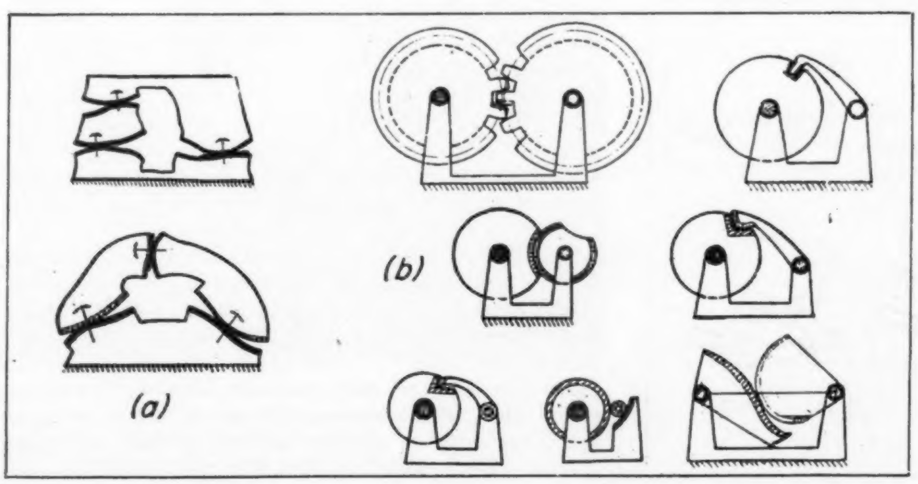


Fig. 17—At *a*, generalized form of all 3-link plane mechanisms, according to Sieker, along with particular 3-link mechanisms, *b*, obtained by giving special forms to the joints

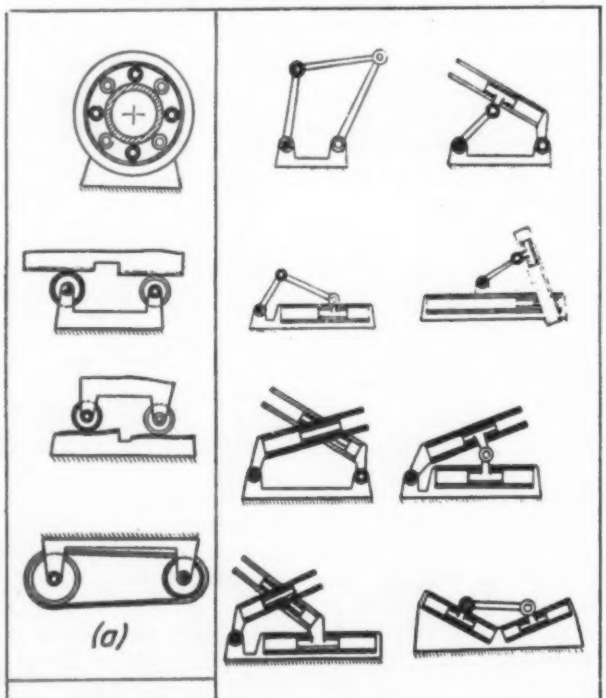
The idea involved here seems to have considerable merit. If carried further, extended to three-dimensional mechanisms, and more examples explained, it might well become generally accepted as the unifying principle tying together the multitude of possible mechanisms, and forming the basis for an improved classification system.

Conclusion: As a background or framework for understanding and study of mechanisms we need (1) an adequate definition of a mechanism, (2) a classification or grouping of the things we have included in our definition, and (3) a unifying principle tying together and showing degrees of relationship between mechanisms.

We might compare ourselves to the biologist, who (1) defines his field to include all living things, (2) assigns all known living things to major classes and subclasses in accordance with certain distinguishing characteristics, and (3) finds a unifying principle in the theory of evolution.

As the matter stands today we have some alternative suggestions to choose from. Speaking for myself, I like Franke's broadening of the definition to include those useful mechanical devices which are only partially constrained, but would not include such things as purely electrical systems. I also like the major divisions in Franke's classification system. In picturing the relationship of various mechanisms to each other I like the line of thought put forth by Sieker. The ideas of these men, as well as the highly respected work of Reuleaux, are worthy of consideration, if only for the stimulation and refreshment they provide.

Fig. 18—Groups of 4-link plane mechanisms, *a* and *b*, and the general 4-link, 4-roll-joint, combination *c*, from which they are derived, according to Sieker



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Recognition and Treatment of Acceleration

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THERE are three phases, in general, in the design of a mechanism. The first phase is the determination of the dimensions of a satisfactory linkage system to give a necessary output motion. The second phase can be considered as the dynamic analysis where the effect of the applied and inertia forces is considered, if such are large enough. The third phase can be considered as the design or proportioning of parts. This paper will be directed at the second phase, the dynamic analysis, with a brief discussion of the effects or inertia forces, with emphasis on the methods of determination of accelerations which is a necessary part of dynamic analysis.

The subject of acceleration is a peculiar one because, by itself, it means very little. Tied in with the mass of a body or with the mass moment of inertia of a body, by Newton's equations, the result is of extreme importance in our everyday design work, whether the case be a very special one where linear and angular accelerations are zero to give us our so important equations of equilibrium or whether the case be a general one of plane motion. Consequently, accelerations can lead us into many areas in design, such as velocity analysis, stress analysis, applied forces, inertia forces, gyroscopic forces, vibration, balancing of engines, fluid mechanics, to name a few. Acceleration is related to velocity by its definition as the vector rate of change of velocity, while all the other phases come from a direct consideration of Newton's equations. Thus, acceleration analysis is only a preliminary step in any design, and, from the design point of view, I cannot think of accelerations without considering the effects of accelerations.

The consequences of accelerations are very readily seen in machines by broken parts or malfunctioning of the machine. *Fig.* 1 shows a series of enlargements from 16-mm film taken at 400 frames per second showing the action of the driving wheels of a steam locomotive on the rails for one complete revolution of the wheel at a train speed of 66 mph with a slipping speed, because of greased tracks, of 108 mph. The tracks were greased to simulate the conditions pro-

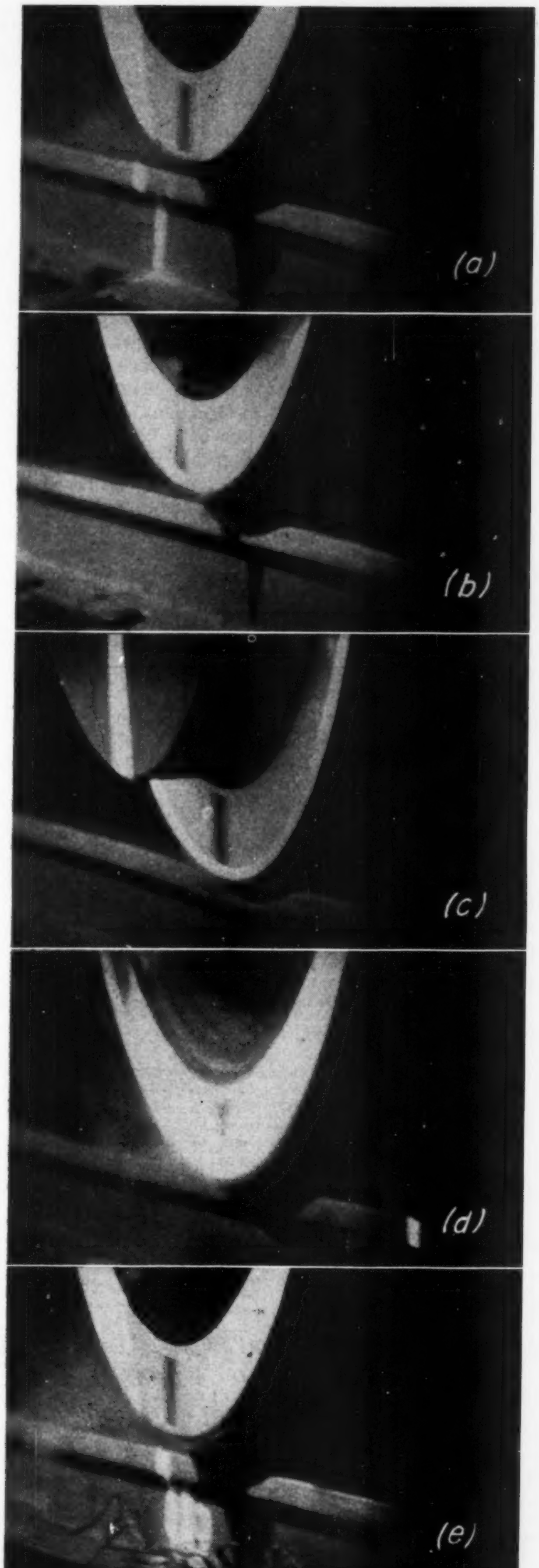


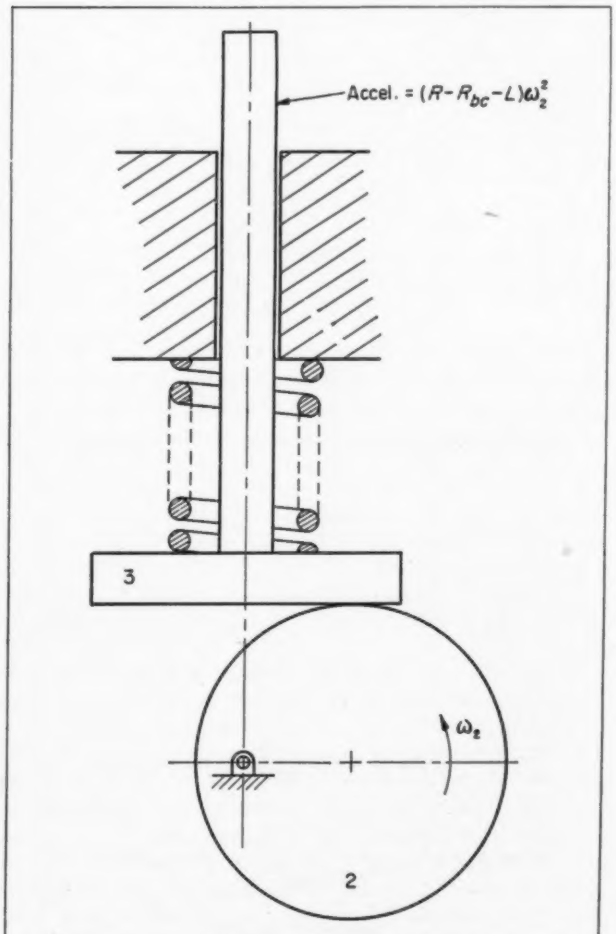
Fig. 1—Action of locomotive wheel at 108 mph, for one complete cycle, from 400-frames-per-second 16-mm film. Photo, courtesy The Timken Roller Bearing Co.

duced by grease on the rails at crossings, frosted and wet rails, or wet leaves on the rails. The tests were made because damage found in some instances did not correspond to what was expected under normal speed conditions. Theoretical considerations indicated that the type of damage encountered could be developed by forced vibrations of the driver wheels at high speeds. Basically, however, the trouble lies in the presence of inertia forces caused by accelerations.

One might suppose that an engine without reciprocating parts, as a constant speed gas turbine, would be free of acceleration considerations. Such is not the case, however, because accelerations are present, plaguing the designer in limited speeds that the material can withstand. Critical speeds, or whirling speeds, with inertia forces at the seat of the trouble, cause concern, as well as unbalanced rotors. It is interesting to note that usually the stresses resulting from the applied loads in gas turbine rotors are very small compared to the stresses resulting from inertia forces.

Fig. 2 is another illustration that will bring out the effects of accelerations. Let us assume that the cam is operating satisfactorily at a given speed. In an effort to increase production, the speed of the engine is increased, with a consequent increase in cam speed. What difficulties might show up? The difficulties are many, such as wearing of the cam, jumping the cam, increased wear on the stem or guide, breaking of the spring. All of the difficulties would perhaps be attributable to accelerations. Replacement

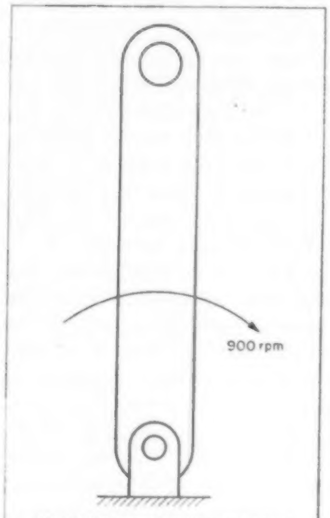
Fig. 2—Cam and follower system



of a broken spring by a heavier spring might take care of the spring problem, but difficulties might be compounded elsewhere as a result.

Fig. 3 shows a simple rod which is rotating at a speed of 900 rpm, let us say. If the speed of the rod is increased to 1200 rpm, how much does the stress increase? Since the acceleration varies as the square of the speed, a 33⅓ per cent increase in speed causes a 78 per cent increase in stress. If the speed is increased to 1800 rpm, the stress is increased four times.

Other illustrations of effects of accelerations might be given by reciprocating engine crank pins wearing out of round because of the difficulty of maintaining proper lubrication with the high inertia forces, noise due to vibration, shaking of engines, poor response in instrumentation because of inertia. The recognition of the results is easy, but the recognition of the problem and the analysis of the problem can be difficult and time taking.



What is the percentage increase in stress if the speed is increased to 1200 rpm?

Fig. 3—Left—A simple but common type of problem

Fig. 4 — Below — An example of mechanisms that cannot be readily "analyzed" on sight

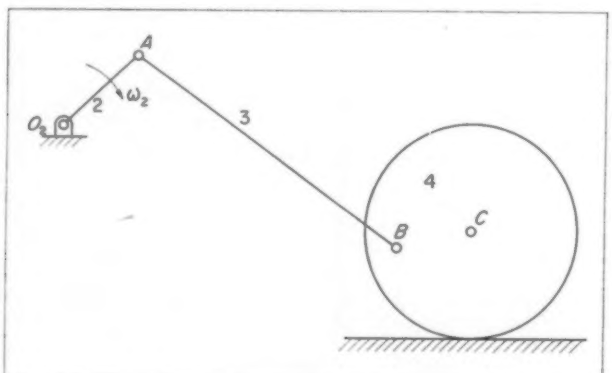
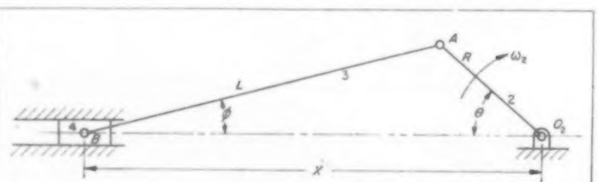


Fig. 5—Slider crank and its analytical equations



$$x = R\cos\theta + L\cos\phi, \text{ with } L\sin\phi = R\sin\theta$$

$$V_B = -R\omega_2\left[\sin\theta + \frac{R}{2L}\frac{\sin 2\theta}{\sqrt{1-(\frac{R}{L}\sin\theta)^2}}\right] \approx -R\omega_2\left[\sin\theta + \frac{R\sin 2\theta}{2L}\right]$$

$$A_B = -R\omega_2^2\left[\cos\theta + \frac{\frac{R}{L}\cos 2\theta\left[1-\left(\frac{R}{L}\right)^2\right]+\left(\frac{R}{L}\right)^3\cos^4\theta}{\left[1-\left(\frac{R}{L}\sin\theta\right)^2\right]^{3/2}}\right] \approx -R\omega_2^2\left(\cos\theta + \frac{R}{L}\cos 2\theta\right)$$

Valid only for ω_2 constant

Our problem, then, in controlling inertia forces and stresses is to control accelerations where possible. And to control accelerations, one must be familiar with acceleration analyses. My job, then, is to review briefly the general principles of acceleration analysis.

Acceleration is a difficult quantity to visualize, because one must be able to visualize both rate of change of magnitude of velocity and rate of change of direction of velocity. Most of us, I believe, sense accelerations by the forces causing accelerations. *Fig.* 4 illustrates a mechanism where link 2 is rotating at a constant speed clockwise. Link 4 is a gear rolling on a rack. What is the direction of the angular speed of link 4? If the motion is apparent, what is the direction of the angular acceleration of link 4? Is it a large or small quantity? Can it be neglected in the design of the system? The feeling of forces can be missing in handling such a problem. Accordingly, the only procedure is the direct application of the mathematical equations used to define accelerations, which requires, also, the definition of velocities. The definitions are the basis of any acceleration solution, whatever the technique used.

$$V = \lim_{\Delta t \to 0} \frac{\Delta s}{\Delta t}$$

$$A = \lim_{\Delta V \to 0} \frac{\Delta V}{\Delta t}$$

where both displacement s and velocity V are vectors. If the vector representation is to be avoided, then the velocity and acceleration can be defined in terms of the projection on two rectangular coordinate axes:

$$V_x = \frac{dx}{dt}$$

$$V_y = \frac{dy}{dt}$$

with the total velocity given by

$$V = \left[\left(\frac{dx}{dt}\right)^2 + \left(\frac{dy}{dt}\right)^2\right]^{1/2}$$

Similarly,

$$A_x = \frac{dV_x}{dt}$$

$$A_y = \frac{dV_y}{dt}$$

with the total acceleration given by

$$A = \left[\left(\frac{dV_x}{dt}\right)^2 + \left(\frac{dV_y}{dt}\right)^2\right]^{1/2}$$

There are two relations necessary to define angular velocity and angular acceleration:

$$\omega = \frac{d\theta}{dt}$$

$$\alpha = \frac{d\omega}{dt}$$

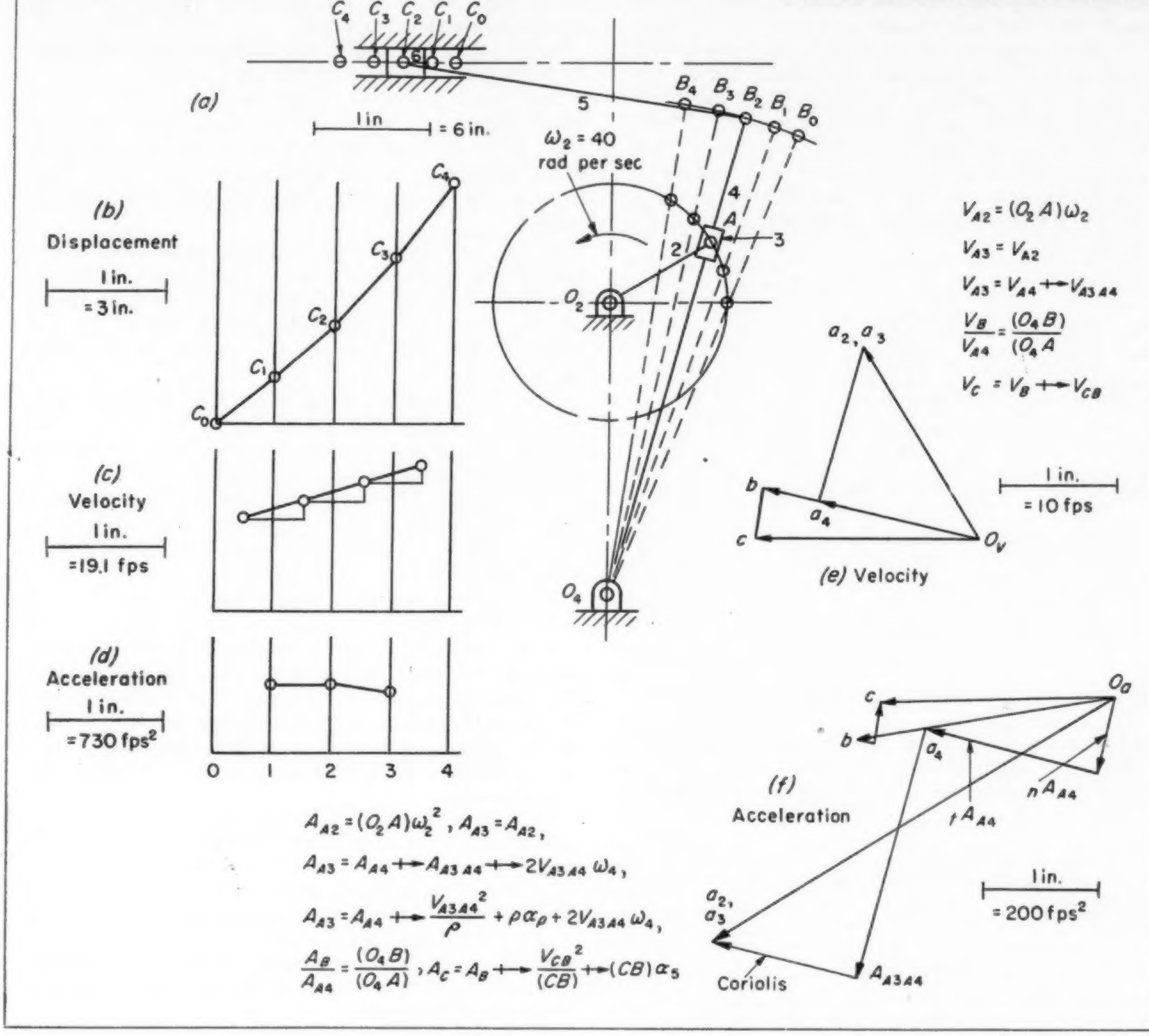
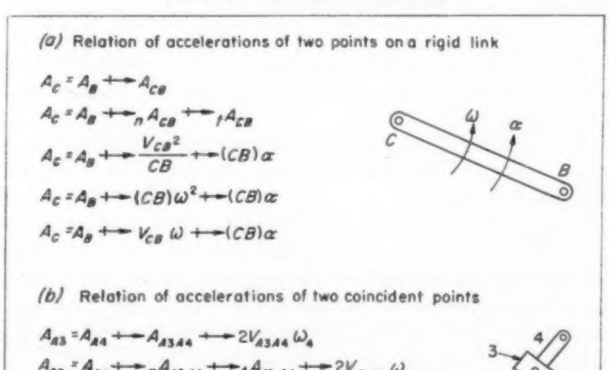


Fig. 6—Above—Analysis of a shaper mechanism. Steps *a* through *d* show method of graphical differentiation. At *e* and *f* velocity and acceleration polygons are shown

Fig. 7—Below—Different ways of writing vector equations for two common situations



There are three basic methods by which an acceleration analysis can be carried out for any given problem. Each will be discussed.

Setting Up Analytical Equations: An equation can be set up for the displacement of a particular point, using rectangular coordinates. Differentiation twice with respect to time gives the acceleration components, which can be combined to give the resultant acceleration. Such a procedure has been set up for a few mechanisms, as the slider crank and various cam arrangements, but the equations for accelerations can become so unwieldy that usually other techniques are used. *Fig.* 5 shows the equations for a slider crank, with approximations usually used in analytical work.

Graphical Differentiation: A technique which requires a minimum of background is the graphical differentiation of the displacement. *Fig.* 6 illustrates the procedure. A shaper mechanism, taken for illustrative purposes, is shown in a particular position for which the acceleration of the slider is desired. The crank is assumed to be rotating at a constant speed of 40 rad per sec counterclockwise. The position of point C is determined for 15-degree intervals of rotation of the crank in *Fig. 6a*, and a plot of displacement-time is made in *Fig. 6b*, with each subdivision of the abscissa

representing $15(2\pi)/360(40) = \pi/480$ seconds. The displacement of point C on the actual layout is plotted to a scale of 1 inch = 3 inches = ¼ foot. The velocity-time curve is obtained in *Fig. 6c* by taking the change of distance in a time interval, considering the change of distance as proportional to the slope of the curve or velocity, and plotting. The curve is magnified twice. By repetition of the procedure, the acceleration curve is determined from the velocity curve and is shown in *Fig. 6d*. The acceleration curve is magnified four times. The scales for the velocity and acceleration curves are given by

$$k_v = \frac{k_s}{n(\Delta t)}$$

$$= \frac{\frac{1}{4}}{2\frac{\pi}{180}}$$

$$= 19.1 \text{ ft per sec}$$

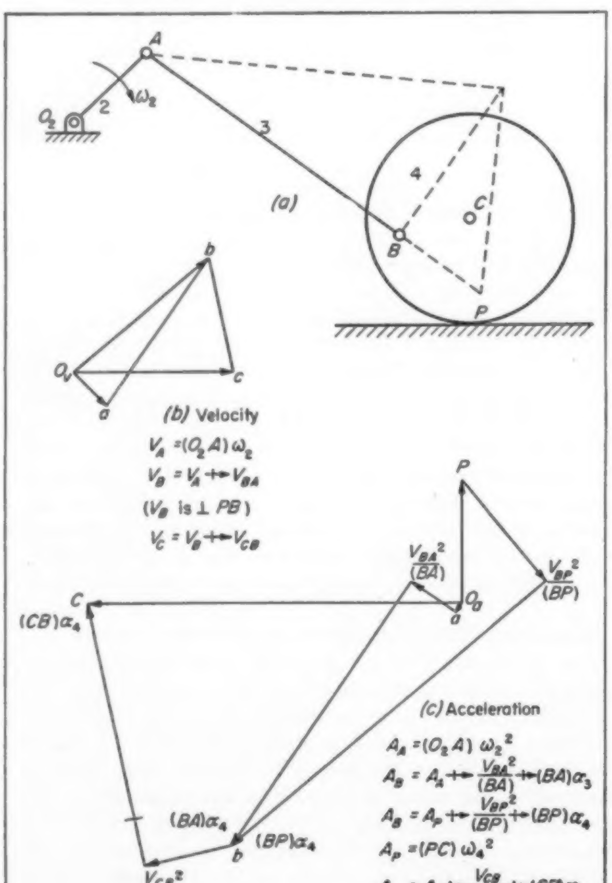
$$k_a = \frac{k_v}{m(\Delta t)}$$

$$= \frac{19.1}{4\frac{\pi}{180}}$$

$$= 730 \text{ ft per sec}^2$$

where k_s is the space scale, n is the magnification factor from the displacement to velocity curve, (Δt) is the time interval between two consecutive positions of the slider, k_v is the velocity scale, k_a is the acceleration scale, and m is the magnification factor from the velocity to acceleration curve.

Fig. 8—Details of velocity and acceleration analysis of mechanism shown in *Fig. 4*



Such a procedure is an approximate one, the error decreasing as the time elements are reduced.

The same procedure could be carried out for a point which is not moving in a linear path, such as point B, except that the motion of point B would have to be projected on two rectangular coordinate axes, and the components of acceleration determined graphically.

If the crank were rotating with an angular acceleration, the position of point C would have to be determined for equal time elements, with unequal angles of the crank, to have a single scale factor for the velocity curve and a single scale factor for the acceleration curve.

Also, the acceleration curve could be found by plotting the velocity-time curve found by some other means than graphical differentiation and graphically differentiating the velocity-time curve.

The angular acceleration of any link can be found in a comparable fashion, where the angular position of any link with respect to a fixed line is plotted as a function of time.

For such an approach to any problem, one need not have any knowledge of Coriolis' component of acceleration, or auxiliary points which might be necessary tools if the third technique to be discussed is used. The problem can become a long, drawn-out procedure if the accelerations of the center of gravity as well as the angular acceleration of each link were desired for a complete cycle of operation for a complete dynamic analysis.

Application of Analytical Relations to Graphical Solutions: A third technique, which requires more background, but which gives neat solutions, is the application of analytical relations to graphical solutions. The tools used are the relative velocity and relative acceleration equations which afford graphical solutions, from which, if desired, analytical relations may be obtained.

The relative velocity equation, as $V_B = V_C +\!\!\to V_{BC}$ or $V_{A3} = V_{A4} +\!\!\to V_{A3A4}$, is the basis of the velocity polygon in *Fig. 6e*, with the equations given in the order used in the solution.

The relative acceleration equation can be expressed differently depending upon the situation. If two points on a rigid link are considered, the relation of accelerations can be given by any one of the forms appearing in *Fig. 7a*.

All the equations are identical in results, with variations in expressing the relative normal and tangential components.

The relation of acceleration of two coincident points, as A_3 and A_4, can be expressed by any one of the forms in *Fig. 7b*.

By application of the relative acceleration equation, as shown with the acceleration polygon in *Fig. 6f*, the complete acceleration polygon can be drawn, from which the acceleration of any point can be determined quickly, as well as the angular acceleration of each link.

Fig. 8 shows the acceleration polygon for the mechanism shown previously in *Fig. 2*. The angular velocity of link 4 is clockwise, with the acceleration of the center of the gear, link 4, being accelerated to the left. Thus, the angular acceleration of the center of the gear is counterclockwise. The acceleration of point A is relatively small compared to the acceleration of points B and C. The inertia forces of links 3

and 4 might be sufficiently large to affect a force analysis considerably.

As mentioned before, analytical relations could be derived for the acceleration of a point on the basis of the velocity and acceleration solutions. In addition, geometric properties, as the instantaneous radius of curvature of the path of a point in a mechanism, can be found from $R = V^2/{}_nA$ where ${}_nA$ is the component of acceleration perpendicular to the velocity.

The dynamic analysis, which utilizes the relations $\Sigma F = MA_g$ and $\Sigma T = I\alpha$ to locate the inertia forces at a distance $h = I\alpha/MA_g$ from the center of gravity of a link, is beyond the scope of this paper but should be considered in the analysis of a machine.

Designers of yesteryear had a relatively simple job with slow-speed engines, where velocities, or the first derivative, were the primary concern. Our problem in design is complicated by the necessity of considering accelerations, or second derivatives. Have we reached the limit of analysis necessary? If smoothness of loading is required, as it must be in high-speed engines, we must also consider the rate at which forces are applied, or the third derivative, for which the term "jerk" has been coined. Some work in this area has been done, especially with cams, and I would like to close by suggesting that a future Conference on Mechanisms should include a paper on the subject.

Application and Design of Noncircular Gears

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WHILE the use of noncircular gearing for industrial applications is relatively recent, this type of gearing has been exhibited as a "novelty" for some time. In fact, Edwin Fellows used both elliptic gears and "square" internal gears as illustrations in his 1914 patent application for an "Apparatus for Generating and Cutting Irregular Gears." The slow acceptance of noncircular gears by machine designers probably has the following basis:

1. Noncircular gears are more expensive to manufacture than standard gears.
2. There is less information available relating to design, wear, reliability, etc. for noncircular gears as compared to conventional gears.
3. In many cases the designer is not aware of what can be accomplished through the use of noncircular gears.

It is hoped that this paper covering industrial applications can dispel some of the doubts relating to the reliability of these gears. At the same time the applications, along with a discussion of the design of such gears, should lead to an appreciation of the direct solution to many motion problems that can be obtained through noncircular gearing. Basically, noncircular gears are those having their teeth arranged on a pitch line which is not a circle concentric with the axis of gear rotation. Examples shown in *Fig.* 1 illustrate some of the variety of shapes included by this definition.

Speedgraphs: To facilitate explanation of the motion performed by the various examples to be considered, each set of gears will be presented with the corresponding speedgraph, *Fig.* 2. A speedgraph has as ordinates the ratio of the angular displacements of the follower to the driver and as abscissa the angular displacement of the driver. In general, the driver is considered as any member having uniform rotation and making one complete revolution per cycle of the machine. With this understanding, the following facts can be established:

1. The area under the speedgraph between any two ordinates is the angular displacement of the follower when the driver moves through the corresponding values of the abscissa. This relation is true even if the driver motion is not uniform. Certain obvious checks on the accuracy of a speedgraph can always be made for the following conditions: (1) If the follower motion is continuous, the total area under the speedgraph must be some multiple of 360 degrees, and (2) if the follower oscillates every cycle, the total area under the speedgraph must be zero.
2. If the driver motion is uniform, the ordinates of the speedgraph are proportional to the follower velocity and the slope is proportional to the follower acceleration.

Part of the speedgraph in *Fig.* 2 is dotted and the corresponding parts of the theoretic pitch line have been cut off in the actual gears. The limitations on noncircular gearing which make this procedure necessary will be discussed in connection with certain of the applications.

Elliptic Gears: Referring to *Fig.* 3, the particular example illustrated is an early application in the printing industry of the classic elliptic gears. As shown, these gears are utilized in a rotary cutter or sheeter to convert a continuous web of paper into a stream of sheets prior to printing. Depending on the job, the sheeter must be able to supply any one of several standard sheet sizes at each printing speed.

In designing such a sheeter, the following problem must be faced. The pinch rolls controlling web speed are set through change gears to meter an amount of paper past the sheeter corresponding to the sheet size to be printed. The sheeter must run one-to-one with the press (one sheet printed per press cycle) and yet at the instant of cutting, the sheeter must travel at web speed to prevent tearing.

As indicated in the illustration, problem of sheeter speed is solved by introducing a pair of elliptic gears into the sheeter train. These are adjusted to produce the required instantaneous velocity corresponding to web speed while maintaining the required average velocity of the press.

This application of noncircular gears has little to recommend its use over a linkage mechanism since all that is required is speed variation. A recent sheeter design (reported in MACHINE DESIGN, October, 1952) employs an ingenious sliding coupled linkage (*Fig.* 3) to perform the same operation. The linkage mechanism offers two advantages over noncircular gears in this case. First, it is more economical to manufacture in limited quantities and second, it has a smoother range of speed variation. Even though the noncircular gears are more compact and more readily balanced, these advantages are relatively unimportant for this type of machinery.

In any application of this type where there is not one, but a wide range of motions to be covered, linkages are better adapted than *any* gears. Once gears are cut their motion is set, but by adjusting the pivot in the mechanism shown at the bottom of *Fig.* 3, for example, a whole class of motions can be obtained.

Noncircular Gears: The examples shown in *Fig.* 4

*Paper presented by Mr. Albert A. Hess, Miehle Printing Press and Mfg. Co.

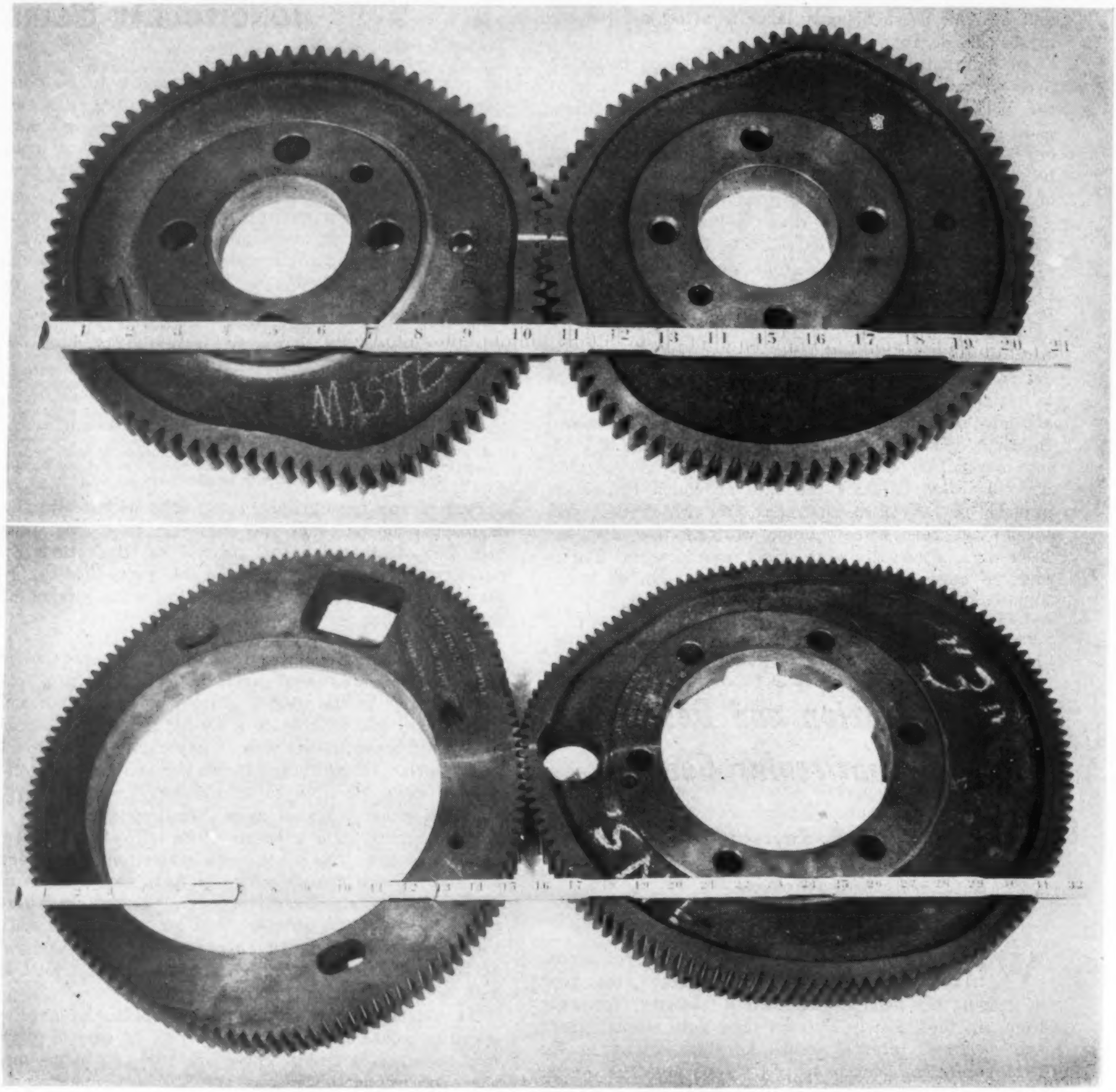


Fig. 1—Master gears used in the manufacture of similar noncircular gears

were produced from master gears by plaster-mold casting. The gear material is bronze; the application is a slow-down mechanism between two presses in tandem. One of the limitations in noncircular gearing is illustrated by these examples; namely, the difficulty of cutting teeth on a cusp. Later gear designs profited from this experience, but these early gears had their pitch lines approximated by circular arcs and straight lines as a manufacturing expedient.

This gearing is designed to catch a sheet traveling at high speed, slow it down and finally urge it gently against fixed stops. While the sheet is stationary it is registered before performing the next printing operation. Specifically, the gears rotate shafts carrying a series of vacuum wheels which contact the sheet when it is released from a chain conveyor. These slow-down wheels are visible through the cut out of the feedboard in *Fig.* 4. As can be seen from the speedgraph, when the slow-down wheels first contact the sheet, they travel at or near conveyor speed to avoid skidding and possible cocking of the sheet. A constant deceleration then occurs until the sheet speed has been reduced to 25 per cent of its original speed. As the sheet is registered and starts into the second press, the suction is cut off and the slow-down wheels are again accelerated to conveyor speed in order to catch the next sheet.

In spite of certain crudities, these gears illustrate some of the advantages of noncircular gears. The basic objective, inertia control of the sheet, is obtained. This is important to prevent skidding or cocking as well as to prevent nicking the leading edge of the sheet as it stops against the front guides. The mechanism itself shares this advantage since the low forces introduced by constant acceleration and deceleration reduce the inertia load on the gears. This leads to a mechanism that is small but adequate for the job; both gears together occupying the space of a man's hand.

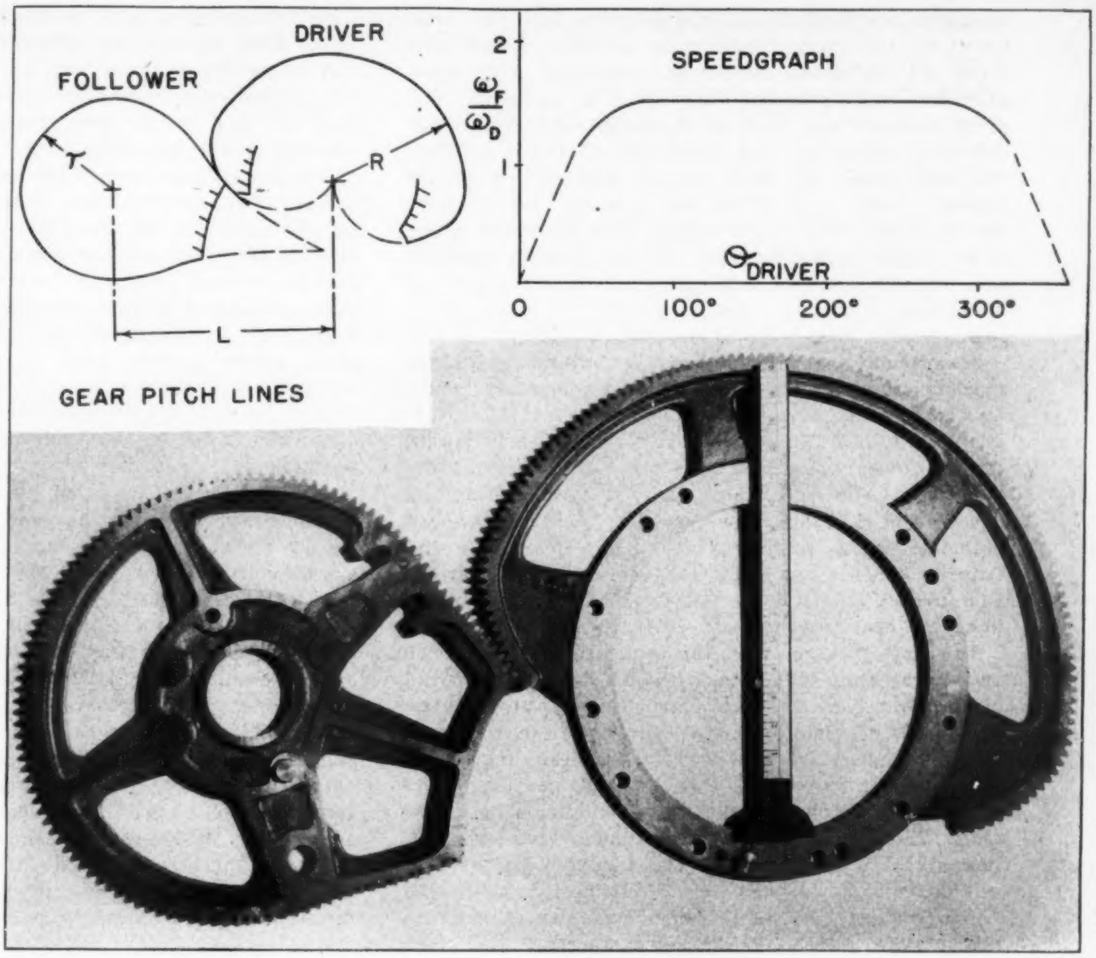
A more compact and equally rugged design employing cams or links is unlikely. In fact, a simple linkage to approximate the sustained low velocity and the

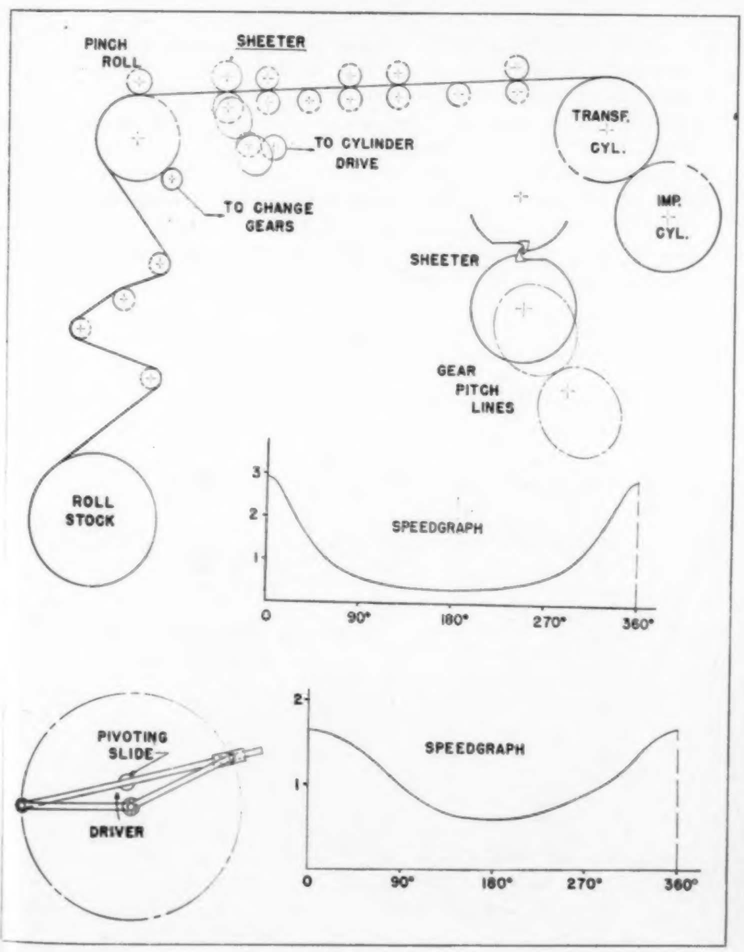
NONCIRCULAR GEARS

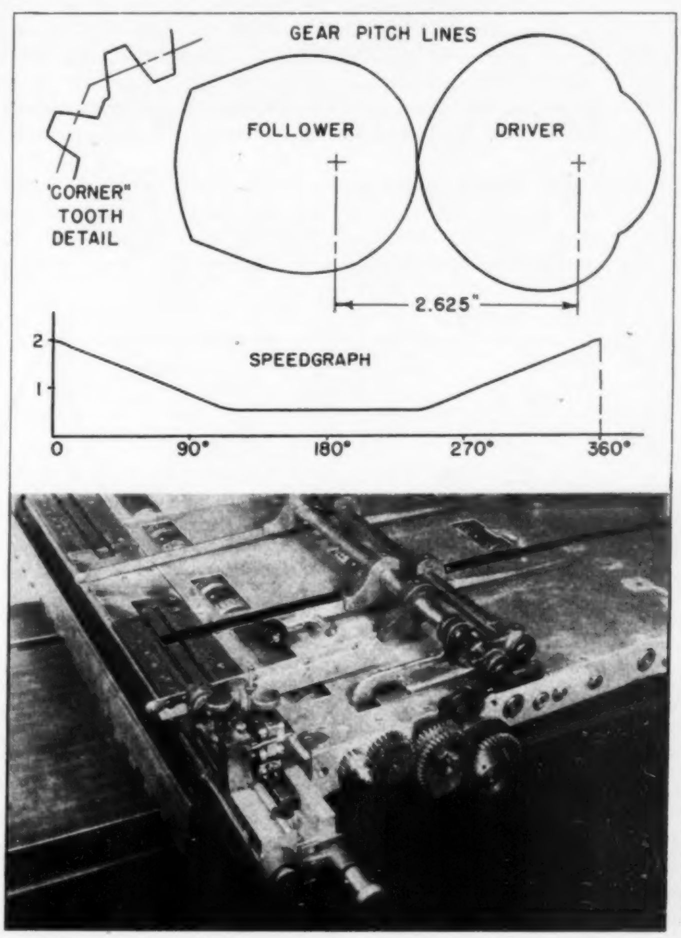
Fig. 2 — Right — One pair of noncircular gears and their associated speedgraph

Fig. 3—Below—Sheeter mechanism and alternate drive schemes

Fig. 4—Below, right—Noncircular sheet slowdown gears and associated speedgraph







constant acceleration and deceleration has not been found as yet. Any combination of cams, gears, and links will introduce additional members with their attendant backlash and any attempt to retain the same compactness leads to formidable difficulty with contact stresses and cam followers. For this application the gears are run without lubricant since oil leakage could result in damaged sheets. Fortunately, the pitch line velocities are low. This particular press is no longer manufactured, but our records indicate over twenty years of continuous service for many installations.

Noncircular Gear Mechanisms: A third printing press mechanism employs a combination of noncircular gears and cams to achieve the motion shown by the speedgraph in *Fig.* 5. This is the drive for a rotary "transfer", a device which acts as a fixed stop for registering a sheet and then as a conveyor to accelerate the sheet from rest and transfer it to a constant-speed cylinder. In the time left after these operations, the transfer quickly completes the remaining half-revolution to act as its own slow-down device and gently bring the next sheet to rest.

The first picture, *a*, in the sequence, *Fig.* 6, shows one of the cam rollers just leaving engagement (end of the first cam stroke on the speedgraph). The use of cams is necessitated in this application by the dwell period required when the sheet is being registered. Any attempt to produce this dwell by gearing would require a tooth on the follower to remain at the driver axis. This undesirable condition holds true even for instantaneous zero velocity and to avoid it, the cam motion extends beyond the dwell period while the corresponding "tail" and "corner" on the mating gears are cut off.

In the second photograph, *b*, the transfer and cylinder are synchronized; the transfer being driven by a true circular segment portion of the noncircular gear. The sheet is transferred at this time and exact motion is required since both the rotary transfer grippers and the cylinder grippers are locked on the sheet for a short period to prevent misregister during transfer.

The humps on the gears which are in contact in *Fig.* 6 *c* account for the constant acceleration and deceleration indicated by the hump in the speedgraph. As mentioned previously, the transfer is being rushed around to arrive ahead of the next sheet.

The final shot of this sequence, *d*, shows a second cam roller engaging to complete the cycle and bring the mechanism to a dwell position. The parabolic bump near the end of the speedgraph is produced by this cam to give a decreasing rate of deceleration when the transfer acts as a slow-down. This particular feature of the mechanism is noteworthy since it allows for variations in the sheet position as an overlapped stream of sheets moves down the feed board. The transfer is fast enough to move in front of a sheet that is ahead of normal position and still gently bring it to rest. At the same time, the decreasing deceleration allows a sheet that is "behind" to gradually overtake the transfer without bouncing into it with a high relative velocity. This is very important since any nicks in the leading edge of a sheet makes register difficult.

Attention is called to the conjugate cams employed in this drive. Wherever accurate motion control is required (during the time when the transfer acts as a stationary front stop in this application) any other form of cam is unsatisfactory for printing machinery. The use of dual cams and followers allows one roller to be preloaded against the other and all backlash can be eliminated, effectively welding the driver and follower into one member.

The final example to be considered is the press drive shown in *Fig.* 7. This drive is used to impart a reciprocating motion to a printing press bed. The full potentential as well as the limitations of noncircular gearing are evident in this application. Again both cams and gears are necessary in this drive; the cams playing an even greater part than in the previously discussed mechanism. In this case, while the driving gear makes one full revolution the follower moves through slightly more than 383 degrees. *Fig.* 8 shows a theoretical pitch line for the print stroke and return stroke gears if no cams were introduced. As can be seen, the profiles are physically possible only if the gears occupy more than one plane—both the driver and follower would have to be composed of layers of gearing. A second limitation that would occur in these gears and must be considered in *any* noncircular gear design is the question of unmeshing. This tendency occurs whenever the angle between the polar radius of the curve and the normal to the curve make an angle exceeding 45 degrees on *both* the driver and the follower.

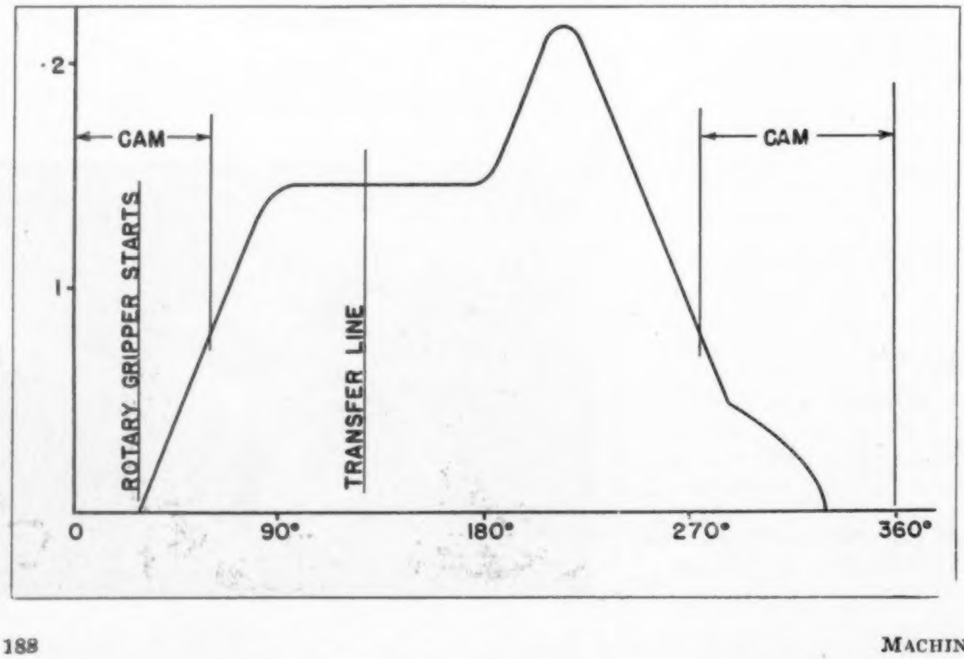
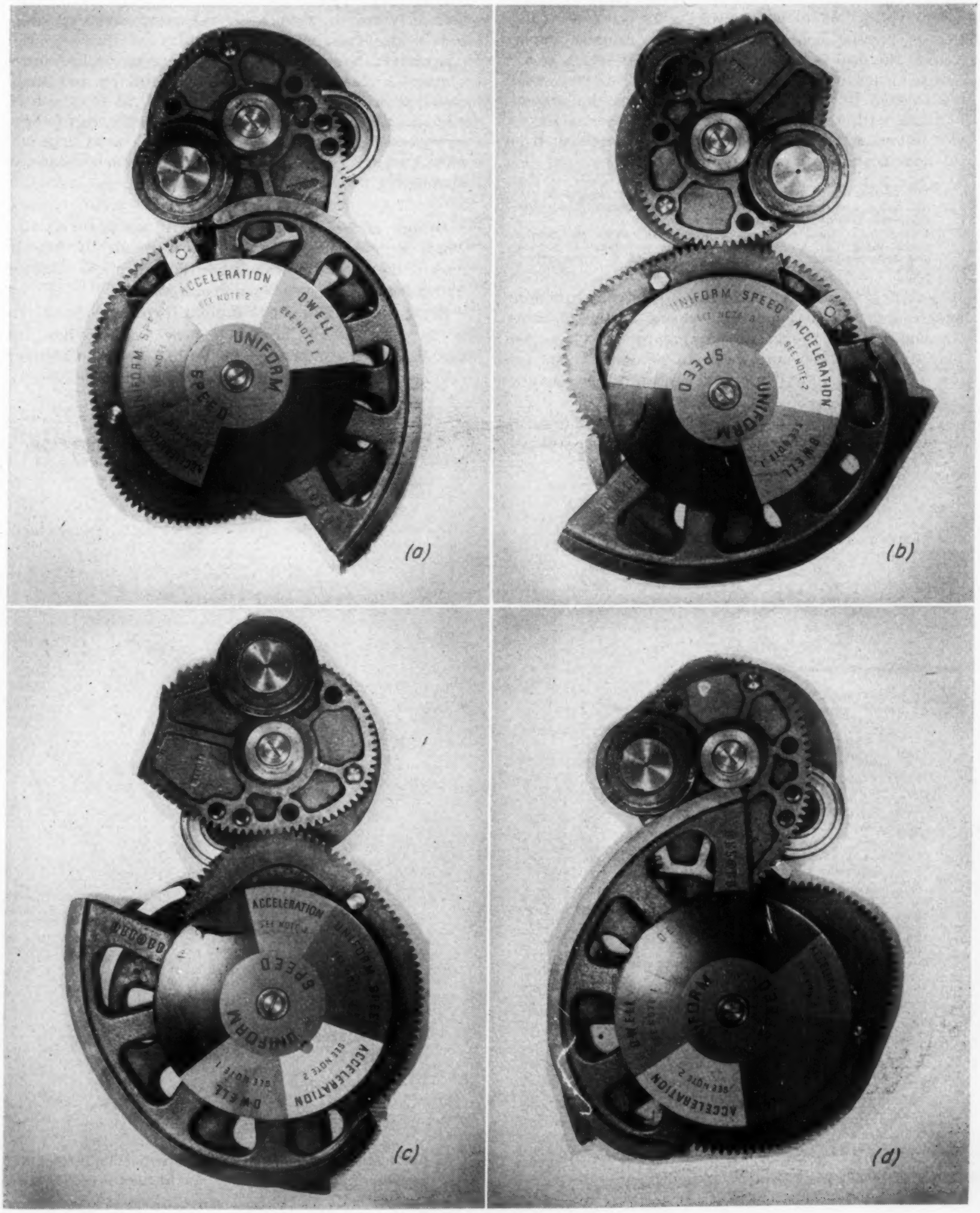


Fig. 5 — Rotary Transfer drive speedgraph

The limitation previously discussed in connection with zero velocity would be present in the theoretical profiles along with the difficulties of cutting teeth in cusps. These Gordian problems of gear design are all unknotted by the time honored method of introducing a cut. The actual gears show what remains of the theoretical profile when the cutting process is complete and conjugate cams are introduced.

A comparison of the motion produced by this drive and the motion produced by an analogous but simplified mechanism, *Fig.* 9, brings out some of the advantages to be attained by noncircular gearing. The simplified mechanism consists of a circular gear rotating in one direction with arrangements for sliding the gear on its shaft to alternately contact either an upper or lower rack. This imparts reciprocating motion to the bed of a press. At either end of the stroke,

Fig. 6—Rotary transfer mechanism shown (a) coming up to speed, (b) at uniform speed, (c) at peak acceleration, and (d) at variable deceleration period



the gear runs out of the rack while a roller engages a straight cam face forming the closing connection between the racks.

Speedgraphs, *Fig.* 10, and acceleration curves, *Fig.* 11, show what has been gained by more complex gearing. These curves have been prepared so that the linear printing velocity is the same for both cases. All results are expressed in terms of the diameter of the driving gear used to give the uniform velocity. As can be seen, the printing stroke is longer while the maximum inertia force has been decreased 25 per cent by the use of noncircular gearing. Since the weight to be reciprocated amounts to 500 pounds in this application with instantaneous power requirements (peak) of 40 horsepower, a 25 per cent reduction in inertia forces is appreciable. Equally important to the user is the reduced end thrust which means less tendency for the machine to "walk" or less strain transmitted to the building which houses the presses.

Along with the tangible reduction in inertia forces, the following intangible benefits are received from the noncircular gear applications:

1. The mechanism is compact and can be run submerged in oil.
2. All cams and gears are practically self-balanced, reducing the possibility of self-excited vibrations.
3. Forces have been reduced without complicating the mechanism train, i.e., at every position *only* two gears or a cam and follower control the motion directly and no intermediary gears or links are introduced.

Item 3 is of primary importance wherever minimum backlash is a functional requirement of the machine. As pointed out previously, conjugate cams are employed in printing machinery to eliminate cam follower backlash. For the same reason, direct connection between driver and follower is a "must" in such a critical item as a press drive mechanism.

Equivalent mechanisms employing cams and circular gears or segments have been studied with the view of reducing costs by substituting such gearing for the present noncircular gear drive. In the examples studied, peak forces have been increased 25 per cent when the mechanism is confined in a somewhat larger space than that occupied by the present drive. Additional problems in preloading cam followers reduces the apparent simplicity of such drives and our current production still utilizes noncircular gearing.

The service history of these gears has been quite satisfactory. Production on the particular drive illustrated began in 1946 and over seven hundred units are in the field at present. Many of these presses run sixteen hours a day and some gear failures have occurred. However, most of these failures are associated with other malfunctioning or abuse of the press. Perhaps the best testimony as to the efficacy of these noncircular gears can be found in our recent improvement program. All redesign was aimed at components outside the noncircular gear drive.

Design of Noncircular Gears: Having demonstrated what achievements are possible through the use of noncircular gearing, an outline of the steps involved in designing such gearing is in order.

SPEEDGRAPH: The starting point for the gear analysis is the speedgraph discussed previously. When the speedgraph is completely determined by the designer's requirements, nothing more can be said. Fortunately,

Fig. 7—Flat bed printing press drive with noncircular gears

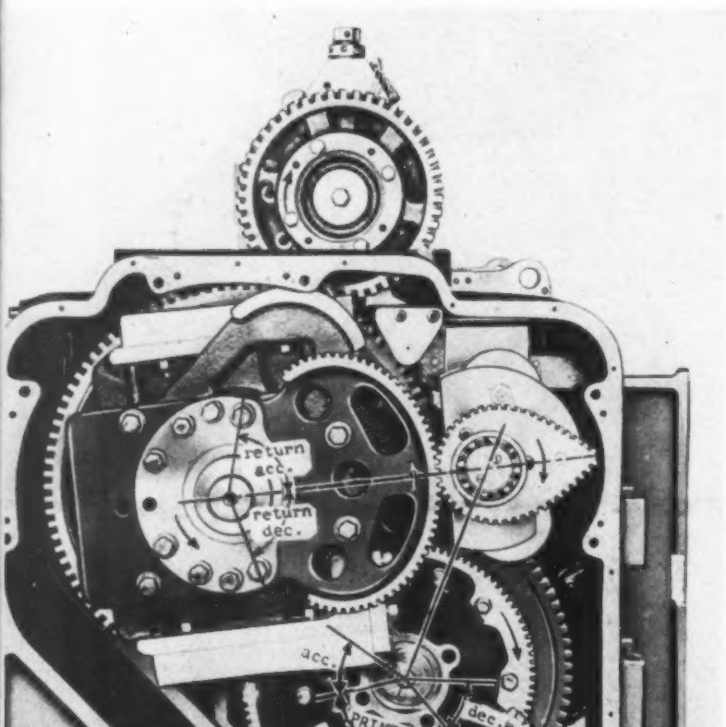
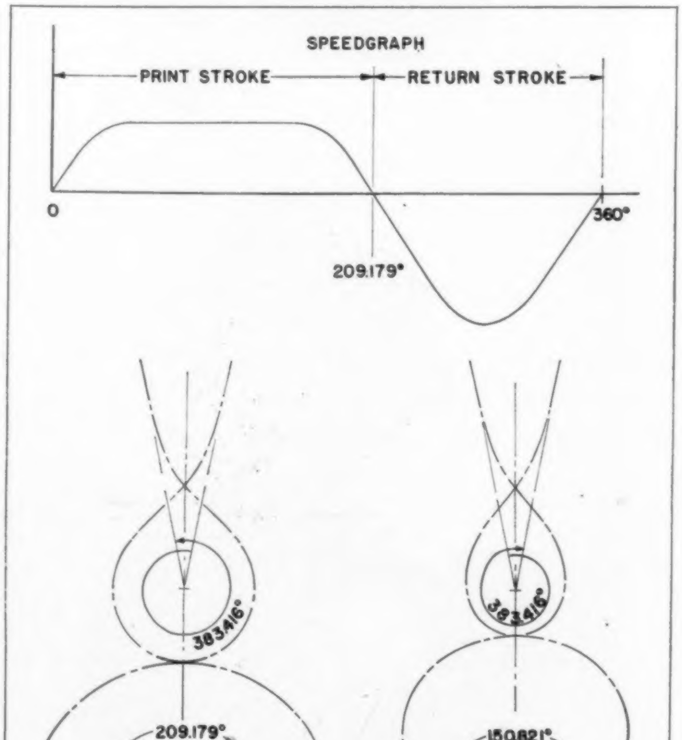


Fig. 8—Gear profiles required to eliminate the cams used in the mechanism of *Fig.* 7



this is generally not the case. Certain parts of a speedgraph will be determined by velocity or displacement requirements of the follower; the remainder of the speedgraph is constructed from displacement and acceleration considerations.

An original speedgraph (dotted line) and its modification are shown in *Fig.* 12 to illustrate this point. In this diagram, the constant-velocity portion in the first half cycle is the primary design requirement. As shown, this part of the speedgraph has been maintained while cutting the maximum acceleration in half. The total area above and below the base line is equal in both cases as is necessary for reciprocating followers. Once the constant-velocity section has been indicated, innumerable curves can be drawn to complete the graph, subject of course to the condition that the total area be zero. By reducing the problem to the selection of a suitable curve rather than to selection of an approximating mechanism, attention is focused upon the primary consideration in high-speed mechanism design. The advantages of the speedgraph approach is that the designer can control the maximum accelerations and then develop the gear or gear-cam combination which will give minimum inertia forces.

Driver and Follower Computations: Once the speedgraph has been established, the next step is to determine the pitch line or rolling cam surface of the gears. This is most readily done in polar co-ordinates since both the radii and angular position are obtained directly from the speedgraph as follows. Let V = the ordinate of the speedgraph at any point, R = polar radius of the driver, r = polar radius of the follower, L = gear center distance, $d\theta$ = angular displacement of the driver, and $d\omega$ = corresponding angular displacement of the follower. Then $R + r = L$ and the pitch lines roll on each other, $Rd\theta = rd\omega$ or $R/r = d\omega/d\theta = V$. Therefore $R = LV/1 + V$ and $r = L/1 + V$. Thus all polar radii are known once the speedgraph is determined and the gear center distance is fixed.

The angular co-ordinates of the driver are fixed by the values of V chosen to determine R and r. The

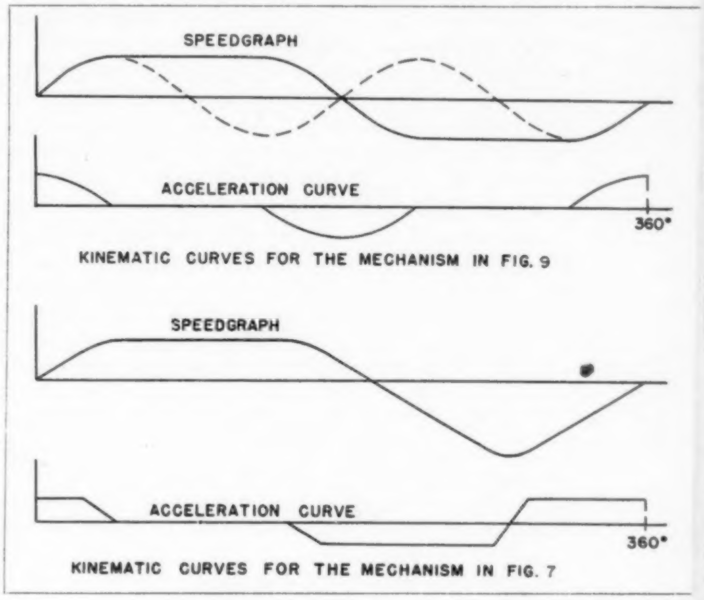


Fig. 10—Above—Speedgraphs for the mechanisms in *Figs.* 7 and 9

Fig. 9—Flat bed printing press drive with shifting circular gears

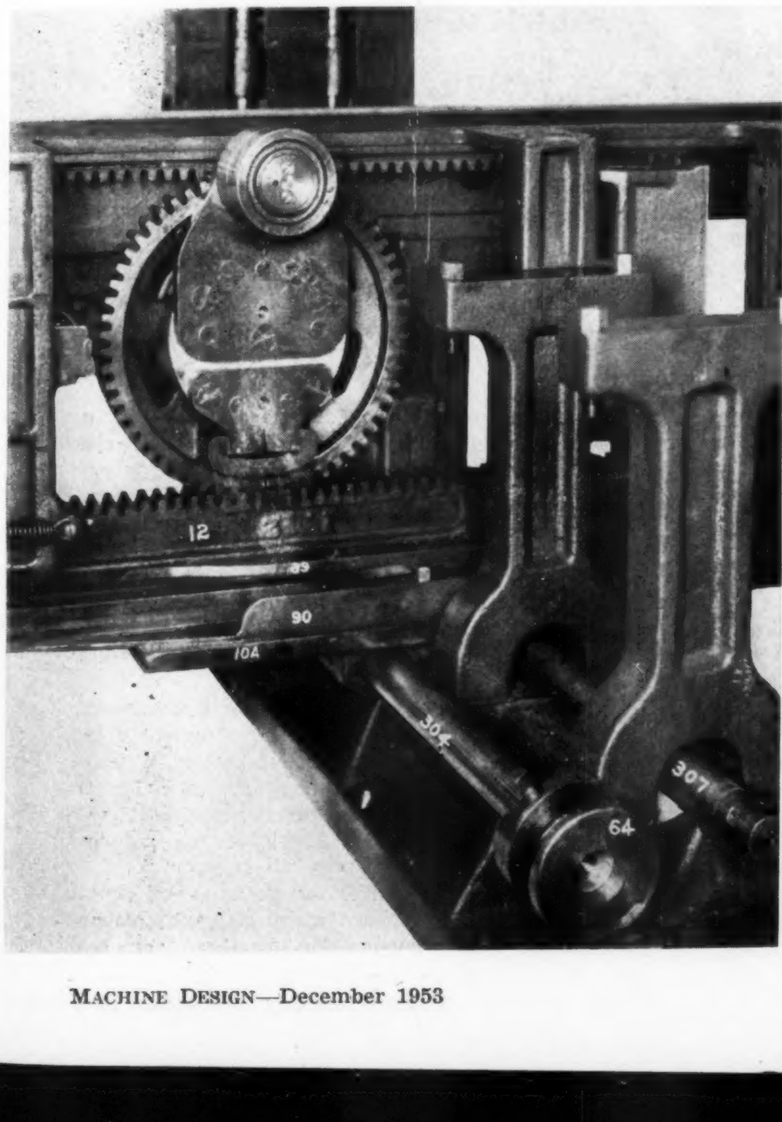
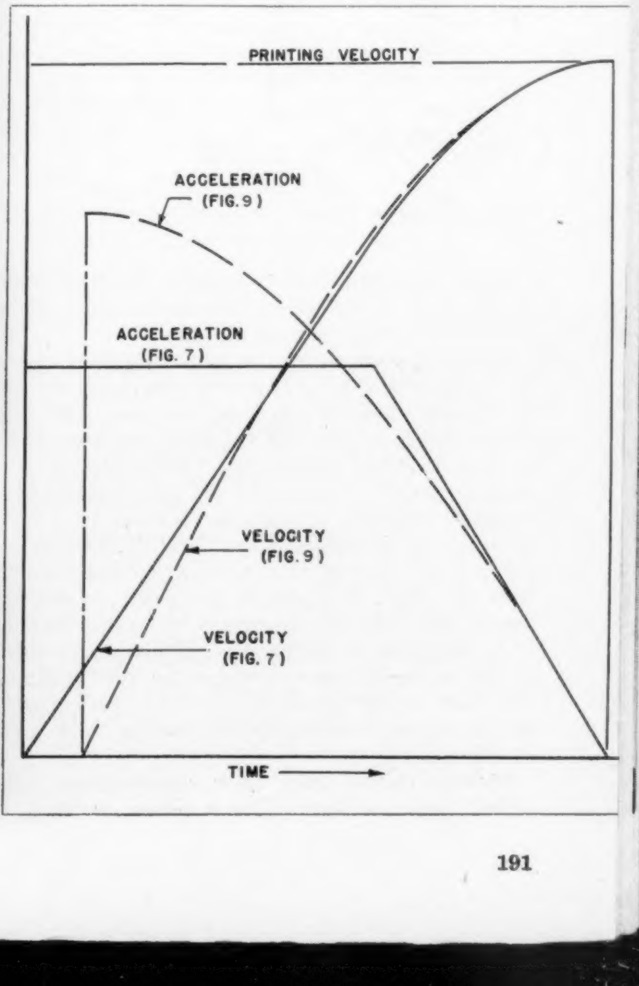


Fig. 11—Below—Velocity and acceleration comparison for the mechanisms in *Figs.* 7 and 9



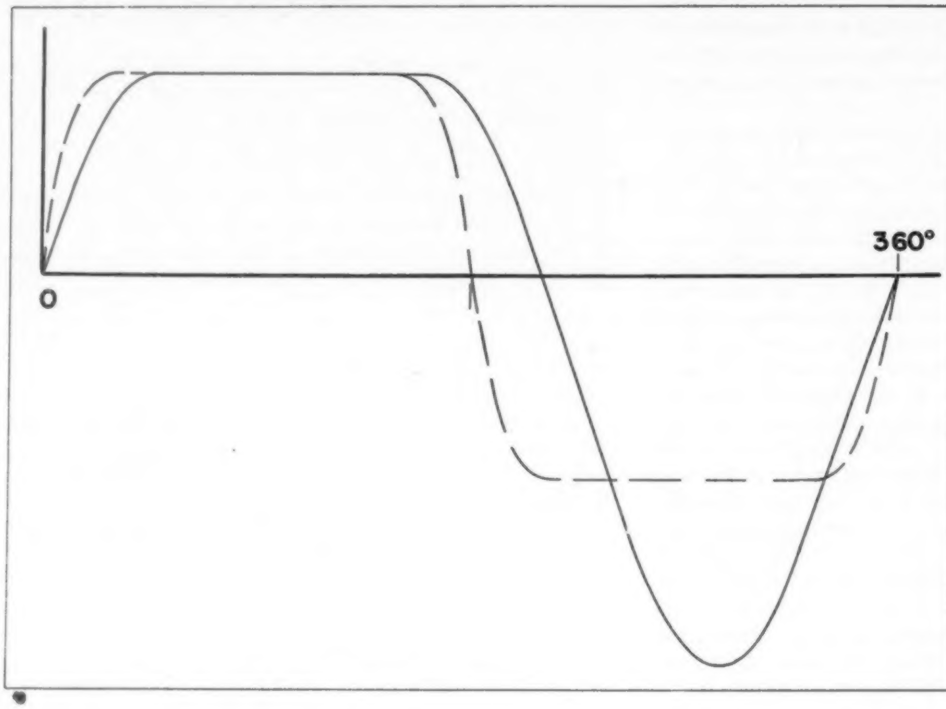


Fig. 12—Designer's speedgraph and improvements. Dotted line shows original and solid line the modified curve

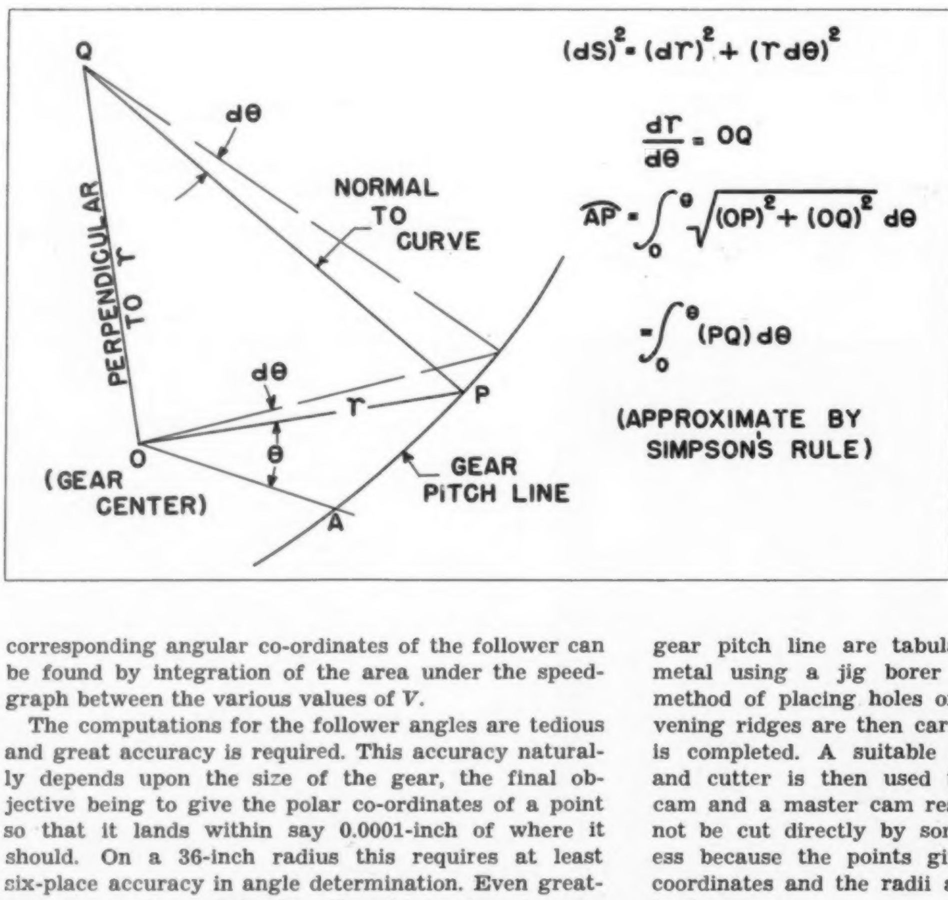


Fig. 13—Approximations for gear arc length computations

corresponding angular co-ordinates of the follower can be found by integration of the area under the speedgraph between the various values of *V*.

The computations for the follower angles are tedious and great accuracy is required. This accuracy naturally depends upon the size of the gear, the final objective being to give the polar co-ordinates of a point so that it lands within say 0.0001-inch of where it should. On a 36-inch radius this requires at least six-place accuracy in angle determination. Even greater accuracy is needed when thought is given to gaging a gear once it is produced. To gage accurately, it is necessary to compute the gear arc lengths precisely, *Fig.* 13, and determine the polar co-ordinates of tooth space locations. Use of Simpson's rule helps in the accurate approximation of arc length, but it has been found advisable to carry all computations to ten decimal places when dealing with a 36-inch radius gear.

MASTER CAMS: Once these computations are completed, the co-ordinates for a series of points on the gear pitch line are tabulated and a cam is cut in metal using a jig borer or some equally accurate method of placing holes on the pitch line. The intervening ridges are then carefully filed and a first cam is completed. A suitable combination of cam roller and cutter is then used to trace this reduced first cam and a master cam results. The master cam cannot be cut directly by some tangential cutting process because the points given are in terms of polar coordinates and the radii are not necessarily normals to the curve.

Naturally, before a complete cam is constructed, the designer must ascertain that the complete gear is usable. Where gearing limitations make it necessary to run off onto supplementary cams, it is pointless to continue the gear pitch line profile. However, the speedgraph will still control the supplementary cam design.

At this stage, the design of the gears is complete. The master cams are utilized in special adaptations of a Fellows Gear Shaper. The problem then be-

longs in the realm of special gear manufacture. Suffice it to say that while such gear production is not as straightforward as the manufacture of commercial gears, it can be done with as much accuracy as required.

Summary: Unfortunately, information on noncircular gear applications and the method of designing is not sufficient to define when noncircular gears should be used. A first general rule, confirming present design practice, is to avoid using such gearing when a design is not penalized by this action. The design and manufacture of such gearing are both time consuming and, therefore, expensive.

However, in those cases where a specific motion is required and a cam or linkage mechanism grows complicated or bulky, noncircular gearing will probably be justified and should be considered. Like all gears, noncircular gears can be separated slightly to allow for manufacturing errors in center distance. This will slightly increase operating backlash but not to the extent required for a similar compensation in a cam mechanism.

It should be noted that where forces are high, noncircular gearing, by maintaining forces in one plane, offers distinct advantages over any linkage mechanism. Basically, noncircular gearing is a powerful tool and like all powerful things should be held in reserve until a worthy situation calls for its use. The underlying mathematics for this type of gearing has been previously covered by Hyman E. Golber and appears in his paper, "Rollcurve Gearing", published in the 1939 Edition of the *ASME Transactions.* A complete understanding of design and manufacturing principles can be acquired by a study of this material.

Utilizing Irregular Gears for Inertia Control

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COMPARATIVELY few people outside of the mining and foundry industries are acquainted with shaker conveyors. This type of conveyor consists of

a long trough or pan line oscillated longitudinally on rollers or ball bearings. The conveyor drive produces a conveying action by accelerating the pan line in a forward direction at a rate having a unit force less than the coefficient of friction of the material and reversing the direction of the pan at an accelerating rate having a unit force greater than the coefficient of friction. The performance of the conveyor depends on the selection and production of definite inertia forces in the pan line and in the conveyed material.

Generally, workable motions have been obtained by rather complex crank driven linkages but, where the crank shaft was rotated at a uniform velocity, the best have fallen short of what was desired. Only by varying the crank shaft speed with an irregular gear was a practical and highly efficient drive produced. *Fig.* 1 shows such a drive with the pan to which the pan line is connected.

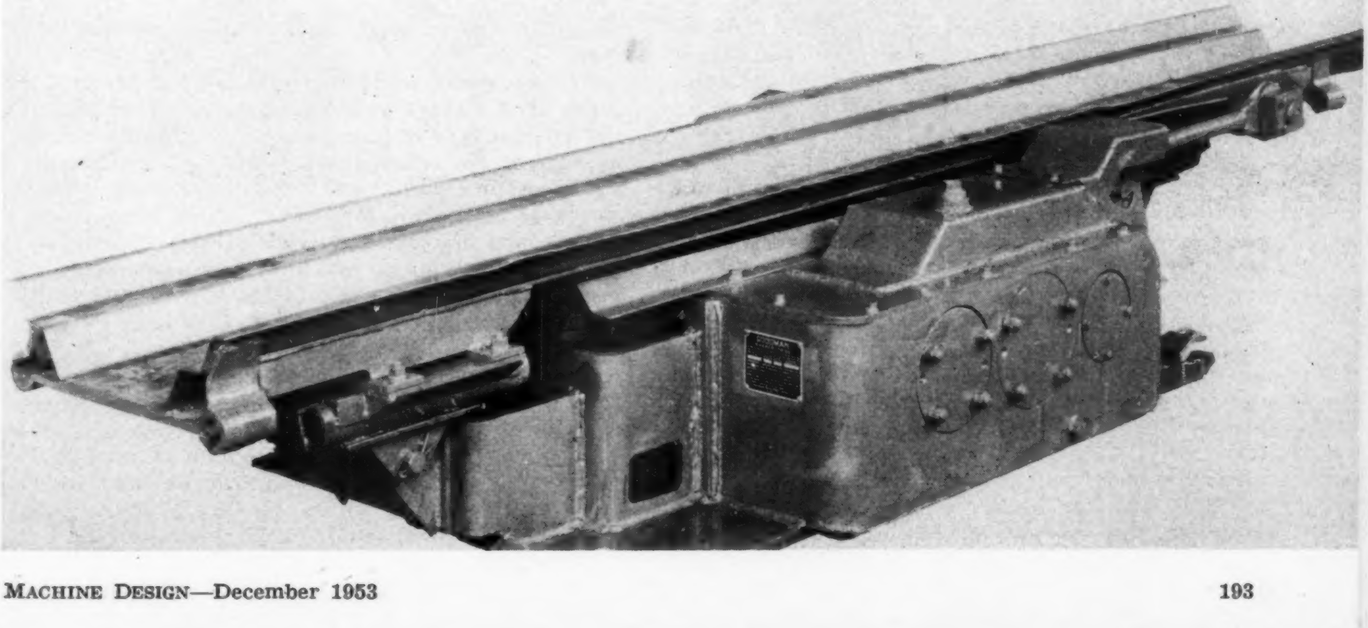
For the purpose of analyzing motions and their conveying ability, use is made of a velocity curve. Such a curve of pan line velocity as produced by a linkage mechanism driven by a constant speed motor with a heavy flywheel is shown in *Fig.* 2. The base of this curve is time, the height is velocity, the area under the curve is the distance of the pan line from the beginning of its stroke, and the slope of the curve, as measured by the tangent of its angle to the base, is a measure of the force of acceleration. The unit acceleration forces are shown by the dotted line.

The area under the curve above the base line shows the position of the pan line on the outward stroke and that below the base line on the inward stroke. A line drawn from the top of the curve to its intersection with the lower part of the curve at an angle whose tangent represents the coefficient of friction of the material, shows the velocity of the material both relative to the ground and to the pan line and the area between this line and the curve is a measure of the material's advance during one cycle.

As the tangent of the end portion of the curve must show an accelerating force less than the coefficient

*Paper presented by L. D. Hagenbook, Chief Engineer, Goodman Mfg. Co.

Fig. 1—Mining shaker conveyor utilizing irregular gears for pan drive mechanism



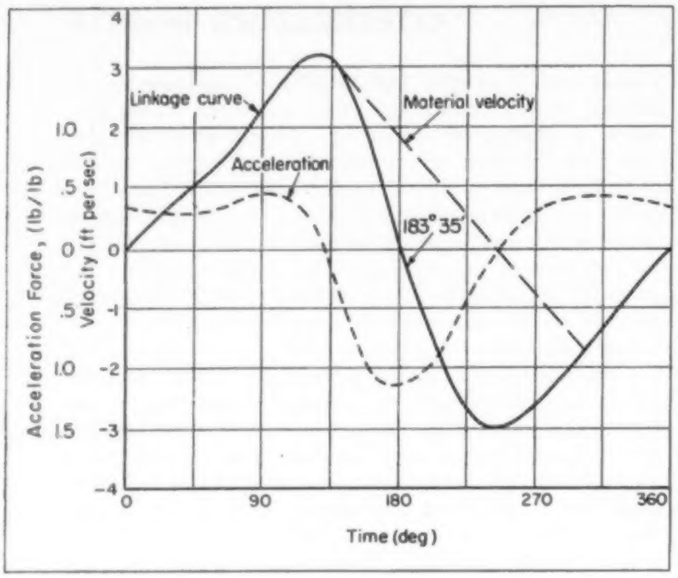


Fig. 2—Velocity and acceleration curves for a shaker linkage with constant-speed drive

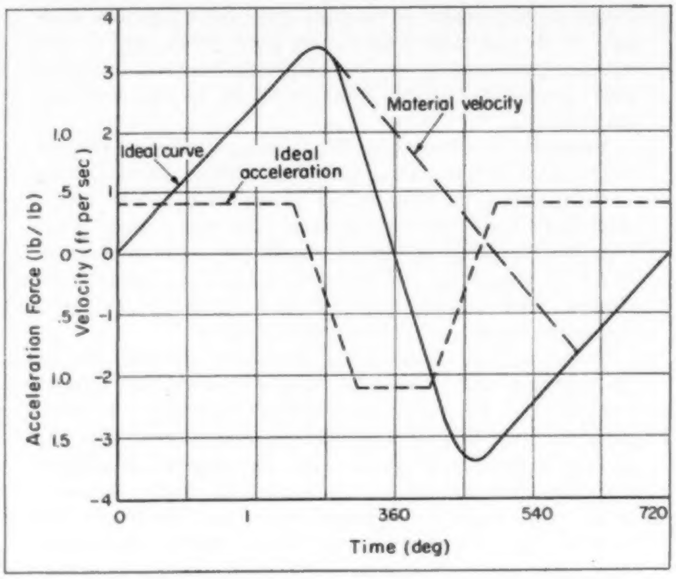


Fig. 3—Velocity and acceleration curves for an ideal drive utilizing uniform acceleration

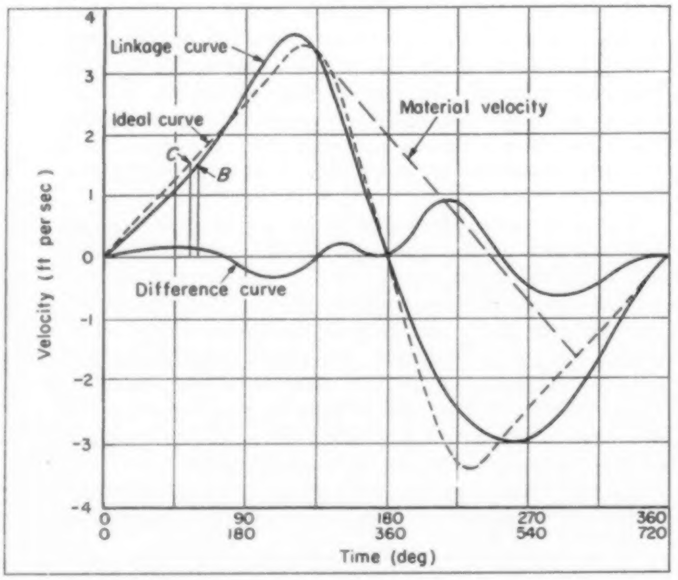


Fig. 4—Velocity curve for a simple linkage system compared with the ideal desired and velocity change curve necessary for linkage modification

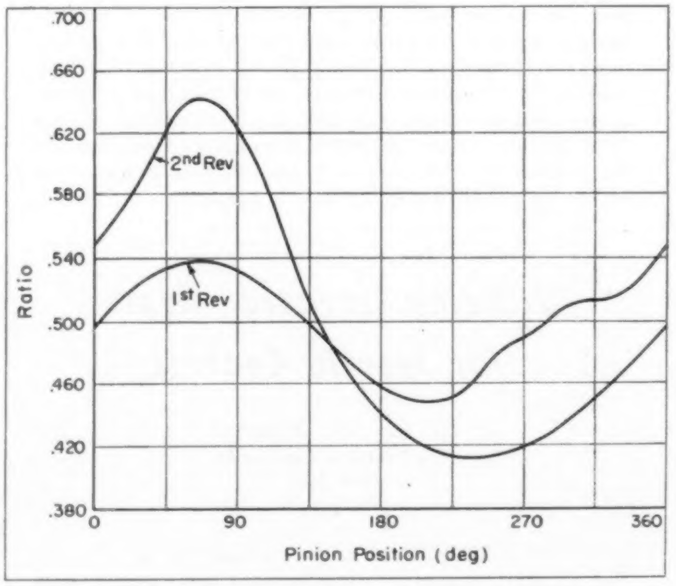


Fig. 5—Plots of gear speeds relative to pinion speeds for a two-to-one drive

of friction, the speed of the drive must be so established that the steepest parts of these portions of the curve show a force within the required limits. The fact that other portions are less steep is of no benefit and actually indicates a loss of time.

In the center of the curve the steepest part represents the maximum inertia stress in the pans and mechanism. From this it is apparent that uniform acceleration permits the fastest operation and the lowest stress for any given conveying ability.

A velocity curve for such a motion is shown in *Fig.* 3 with the unit acceleration forces indicated by the dotted line. To reduce the shock when the driving forces are reversed from one direction to the other the change should be made at a uniform rate. This is shown by the straight sloping lines in the acceleration curve. In the velocity curve these produce parabolas which join the straight lines. Division of the base into 720 degrees is used because the pinion shaft speed is constant and the crank shaft speed is not. The pinion shaft makes two revolutions per cycle.

We have never found it possible to produce such a curve by a linkage mechanism alone, but by taking the simplest type of linkage and correcting its motion by varying the velocity of the driving crank, the desired ideal motion has been exactly obtained. *Fig.* 4 shows, in solid lines, the velocity curve for a simple linkage and in dotted lines, the ideal curve. The wavy line shows the velocity changes necessary to make the linkage motion into the ideal motion.

The speed variations could have been obtained with a pair of irregular gears having an average one-to-one speed ratio but ordinarily a shaker drive has three speed reductions between the motor and crank shaft and to have used a one-to-one set of gears would have added two more gears with an extra shaft and set of bearings. To avoid this, we investigated the possibility of combining the varying speed gears with one reduction and found it possible to do this

with a straight two-to-one reduction in speed.

To design the gears it was necessary to know the required velocity ratios between pinion and gear at their various angular positions. As the velocity of the pinion in constant and it makes two revolutions

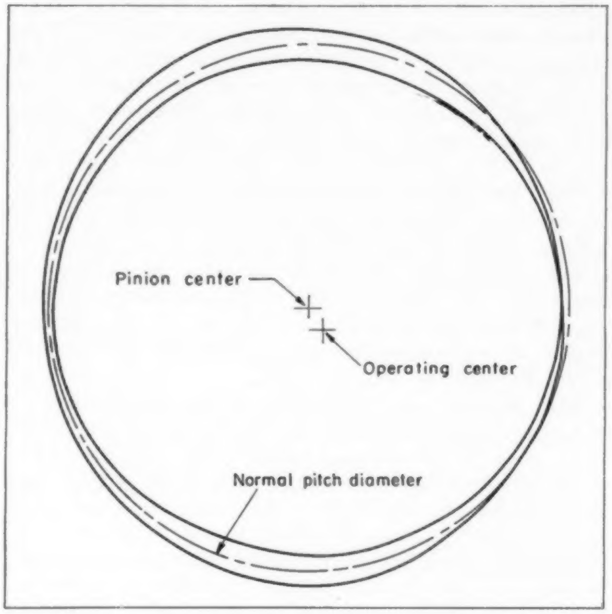


Fig. 6—Above—Plot of pinion centers and normal pitch diameter to be used

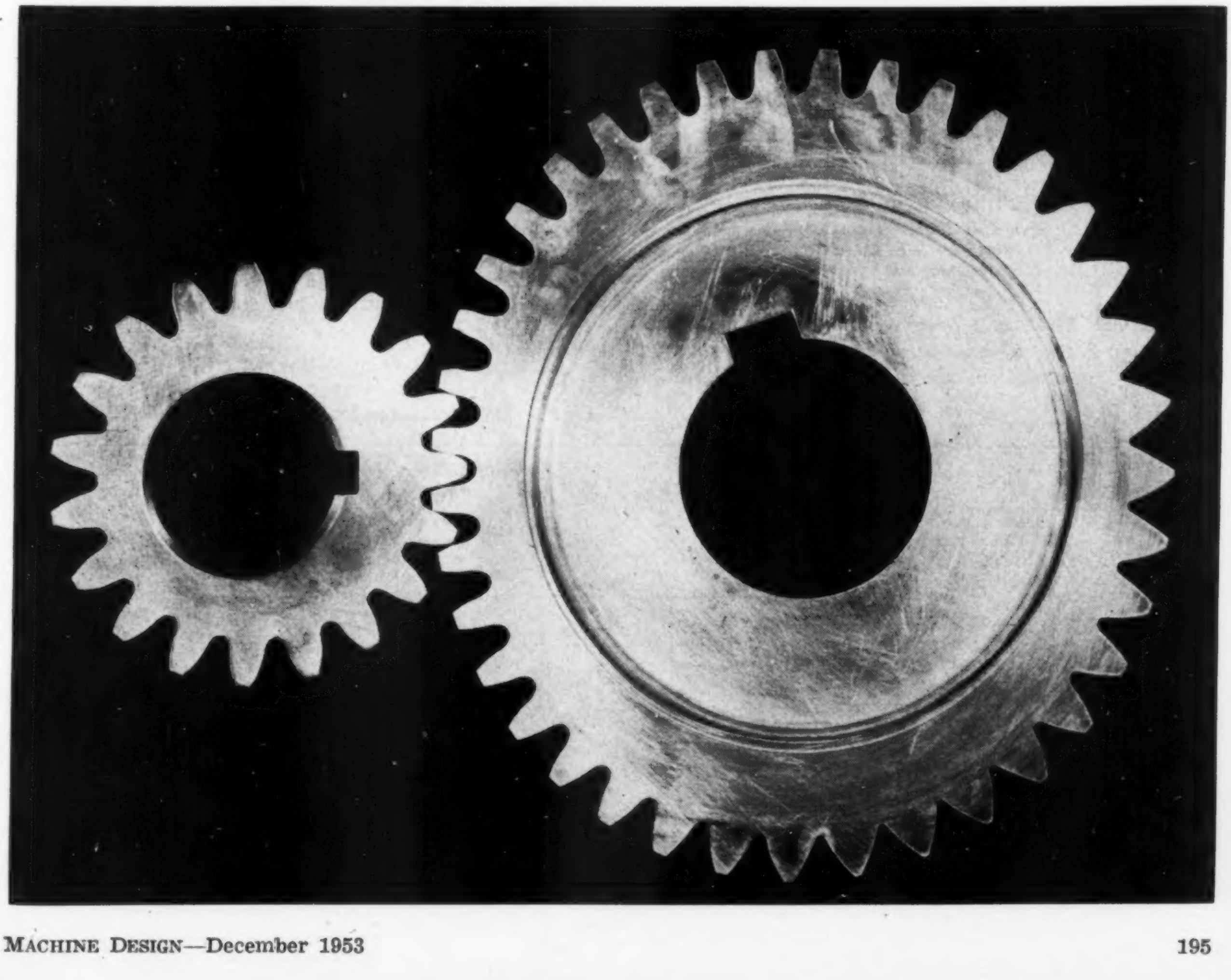
Fig. 7—Below—Final eccentric pinion and irregular gear developed. As much as 60 hp load is normally sustained by this gear drive in regular operation without difficulties

per cycle, any position on the base of the curve will show the position of the pinion if the base is divided into 720 degrees. The area of the ideal curve up to that point will show the corresponding position of the pan line. Such an area under the curve is up to the line at *C*. If a position on the linkage curve having the same area, such as at *B* is located, the distance of this line from the starting position shows the angular position of the gear if the base line is divided into 360 degrees.

The ratio of the velocities at *C* and *B* is the amount by which the average two-to-one speed ratio of the gears is to be altered for the corresponding pinion and gear positions. Determining the corrected speed ratios and plotting the gear speeds relative to the pinion speeds for the two revolutions of the pinion gives the curves as shown in *Fig*. 5.

Gear centers having been established, the pitch radii at which the pinion is to operate are determined and plotted as shown by the solid line in *Fig*. 6. By trial, the radius and center of a circle most nearly approximating this curve is found. This is shown in dot-dash lines. The diameter of this circle is the normal pitch diameter of the pinion to be used. The teeth are to be of standard profile and the pinion is to be mounted eccentrically by an amount equal to the distance between the two centers.

As the gears move, the line of contact is continually changing, but is always tangent to the generating circle of the pinion and passes through the end of the pitch radius on the line of centers. Size of the teeth must be such that the variations in the pitch radius



remain well within the working depth of the tooth and a pressure angle must be chosen that will give adequate tooth strength, no loss of tooth tips and sufficient overlap of tooth contact.

Only a few teeth were in critical positions but when the values for these positions were determined, the profiles for both sides of all the teeth were developed to make sure the gears would operate satisfactorily. The pinion, having a standard involute profile is produced in a Fellows Shaper without any special fixtures except an eccentric arbor.

To produce the gear, an auxiliary arbor is used. By means of cams, this arbor is moved relative to the regular arbor so that while the cutter is rotating at a constant speed the gear blank is rotating at varying speeds of the same pattern that the gear in the shaker drive is to have. The cutter has the same contour as the pinion and is mounted eccentrically. A round gear blank is put in the shaper and the irregular contour produced by an eccentric disc cutter. The disc cutter is then replaced by a toothed cutter and the gear finished. *Fig.* 7 shows the gear and pinion. All the teeth in the pinion are alike but every tooth in the gear is different.

Much more time was required to design this type of gear set than would have been needed to design a pair of one-to-one irregular gears but one pair of gears, a shaft and its bearings have been eliminated and only one irregular gear need be produced instead of two. It is possible to obtain a speed variation of about 20 per cent in addtion to what is obtained from the eccentricity of the pinion. In the gears shown, the highest speed is 56 per cent above the slowest speed. This type of gear and the fixture for producing it are covered by patents owned by Goodman Mfg. Company.

A Simplified Approach to Linkage Design

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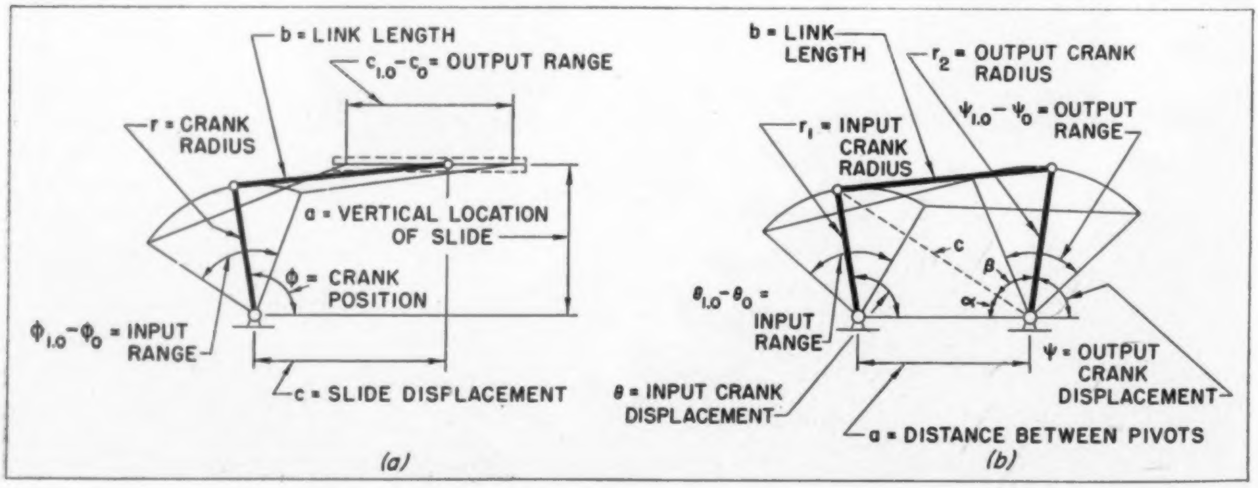
ONE OF THE ever-present problems in machine design is the mechanization of various interrelated motions. These motions are usually preassigned, often quite arbitrarily, and specify the relationships of moving parts or simply the end motion of a single part. Illustrations are present in all types of machinery. Typical examples are seen in textile machinery, packaging machinery, printing presses, valve mechanisms in steam locomotives, machine tools, automotive equipment, household articles, instruments, computing devices and many other common mechanisms. Upon closer observation, it will be noticed that all these devices are simply combinations and arrangements of basic mechanical elements such as: gear trains, cam actuators, cranks and links, sliders, belts and pulleys and other rotating and sliding parts. Combinations of the crank, link and sliding elements are commonly termed bar linkages. These devices are seldom arrived at by pure logical design, but result only from the efforts of specialists. The simplicity or complexity of a mechanism depends wholly upon his experience and ingenuity. We often marvel at the ingenuity of some particularly intricate linkage design. It is just this factor of ingenuity that has limited the use of linkages in machine design.

For purposes of discussion, these motion devices may be defined as either motion generators or function generators. Motion generators produce some particular motion relationships where the output may be restrained to one, two or three degrees of freedom. Often, only the two end points and perhaps some intermediate condition are specified, with the in-between motion having no particular importance. Function generators, on the other hand, have their entire motion specified and are restrained to one-degree of freedom, either translation along a line or rotation about some axis. Mechanically there is no difference between either type. The input-output relationship of a function generator may be specified by an explicit mathematical expression or by some tabular data, whose explicit function is not necessarily known. It is in this latter sense that the function generator will be considered, where the motion desired is not one for which a known linkage exists.

Cams versus Linkages: The most well known solution for a function generator is the cam; flat cams for functions of single variables and barrel cams for functions of two variables. Aside from the kinematics, cam design is very straightforward and consists simply of assigning coordinates corresponding to the specified functions. Perhaps the least known solution is the bar linkage. Both devices have their advantages and limitations.

Fig. 1—Harmonic transformer, *a*, and four-bar linkage, *b*



Cams in general require less design effort and are able to mechanize discontinuous or jagged-shaped functions. Essentially, an infinite number of dimensions determine a cam profile. High accuracy becomes impractical to achieve except at a limited few points. In order to limit backlash, the follower must be spring-loaded against the cam surface. This introduces high friction forces and may limit the acceleration of the driving mechanism.

As computing devices, linkage mechanisms enjoy a

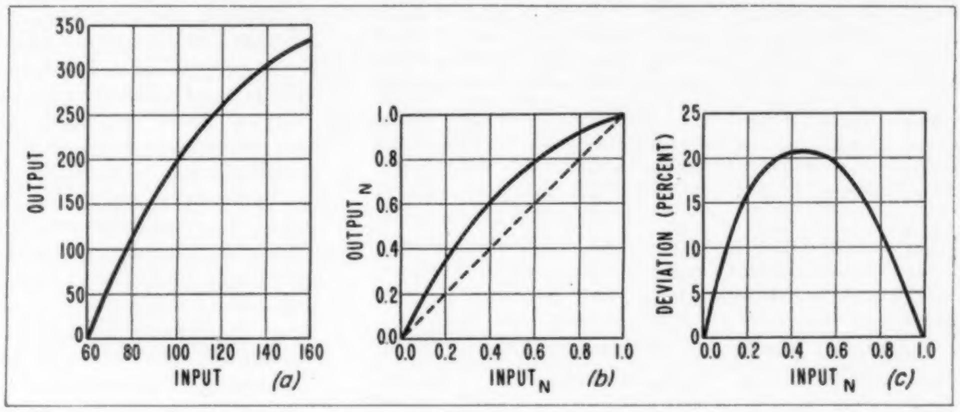


Fig. 2 — Right — Curves for a typical linkage showing: *a*, output plotted against input; *b*, the normalized linkage trace; *c*, the deviation curve

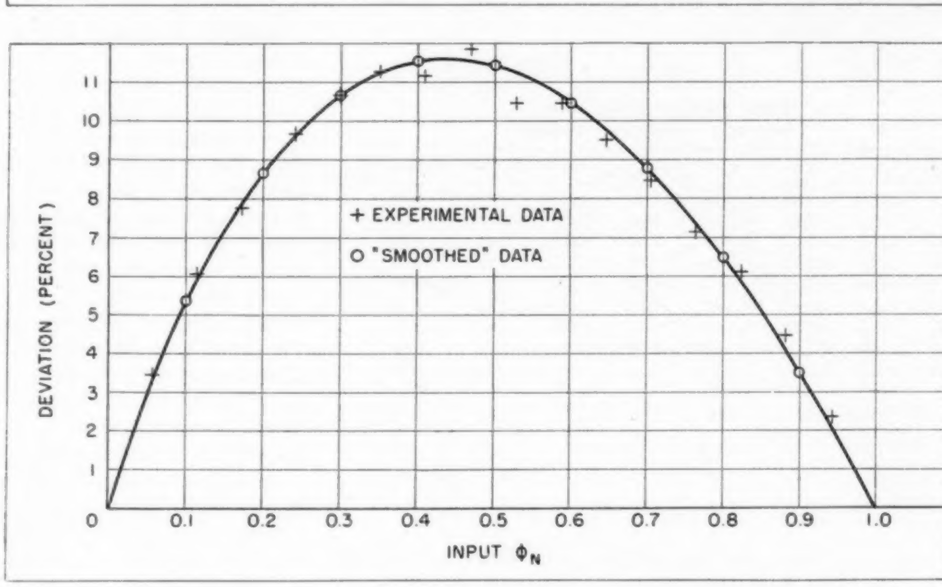


Fig. 3—Right — Required deviation curve for a harmonic transformer

Fig. 4 — Below — Layout of unit harmonic transformer by overlay method

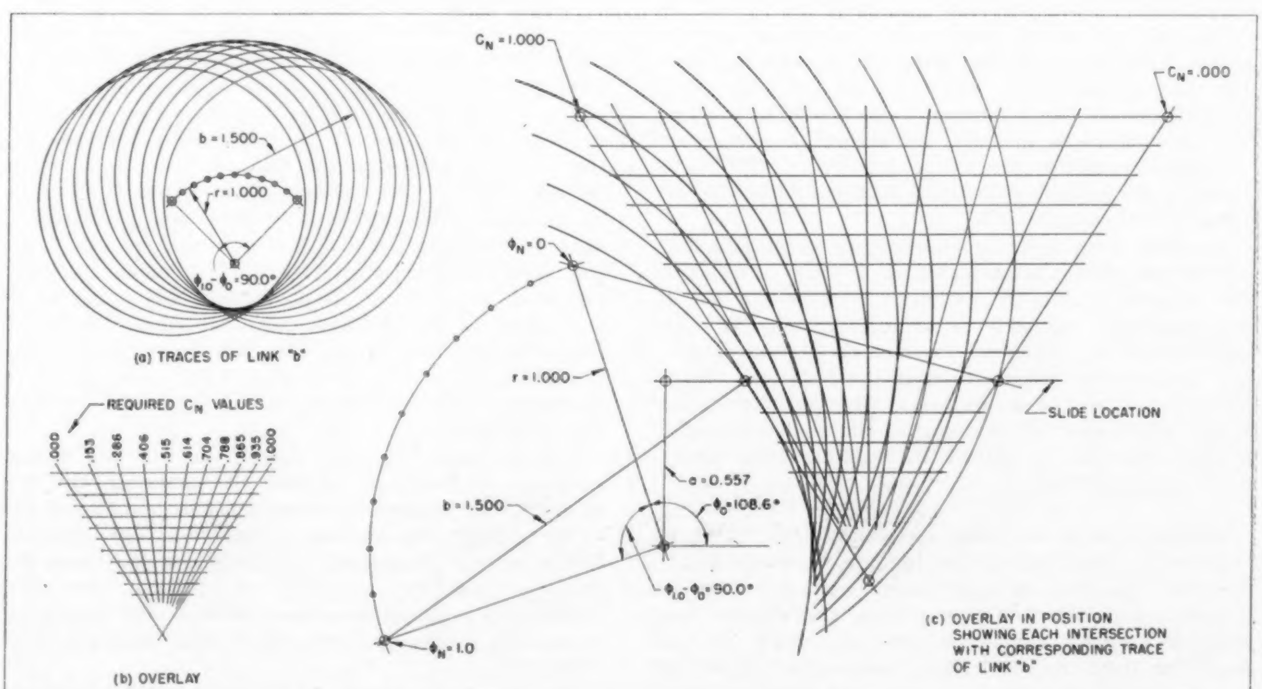


Table 1—Input-Output Requirements for a Harmonic Transformer Linkage

Experimental Data					Linkage Data		
Input*		Output†		Deviation	Input	Dev.	Output§
ϕ	ϕ_N	c	c_N	ε	ϕ_N	ε'	c_N'
1501.5	0.0000	83.335	0.0000	0.00	0.0	0.00	0.0000
1549.8	0.0569	81.578	0.0916	3 47	0.1	5.33	0.1533
1598.2	0.1139	79.988	0.1745	6.06	0.2	8.65	0.2865
1649.2	0.1739[1]	78.507	0.2517[2]	7.78[3]	0.3	10.65	0.4065
1706.9	0.2419	76.842	0.3384	9.65	0.4	11.52	0.5152
1756.5	0.3003	75.537	0.4065	10.62	0.5	11.42	0.6142
1800.0	0.3515	74.427	0.4643	11.28	0.6	10.42	0.7042
1850.0	0.4104	73.319	0.5221	11.17	0.7	8.76	0.7876
1900.0	0.4693	72.055	0.5880	11.87	0.8	6.46	0.8646
1950.0	0.5282	71.198	0.6326	10.44	0.9	3.48	0.9348
2000.0	0.5871	70.070	0.6914	10.43	1.0	0.00	1.0000
2050.0	0.6460	69.120	0.7409	9.49			
2100.0	0.7049	68.189	0.7895	8.46			
2150.0	0.7637	67.318	0.8349	7.12			
2200.0	0.8226	66.385	0.8835	6.09			
2249.4	0.8808	65.503	0.9253	4 45			
2302.2	0.9430	64.792	0.9665	2.35			
2350.6	1.0000	64.150	1.0000	0.00			

*Rotary input required. †Linear output required, total travel to be 0.7553-inch. [1] Normalized Value of ϕ, example: (1649.2—1501.5)/(2350.6—1501.5) = 0.1739. [2] Normalized Value of c, example: (78.507—83.335)/(64.150—83.335) = 0.2517. [3] Deviation, example: (0.2517—0.1739)100 = 7.78.
§$c_N' = (\varepsilon'/100) + \phi_N$.

Table 2—Calculation of Unit Harmonic Transformer Linkage

ϕ_N	ϕ	c*	c_N	ε_{mech}	ε'	Error	Zero Set
0.0	108.6	1.1292	0.0000	0.00	0.00	0.00	0.04
0.1	117.6	1.0001	0.1502	5.02	5.33	—0.31§	—0.27†
0.2	126.6	0.8835	0.2859	8.59	8.65	—0.06	—0.02
0.3	135.6	0.7787	0.4079	10.79	10.65	0.14	0.16
0.4	144.6	0.6847	0.5173	11.73	11.52	0.21	0.25
0.5	153.6	0.6001	0.6157	11.57	11.42	0.15	0.19
0.6	162.6	0.5235	0.7049	10.49	10.42	0.07	0.11
0.7	171.6	0.4533	0.7866	8.66	8.76	—0.10	—0.06
0.8	180.6	0.3886	0.8619	6.19	6.46	—0.27	—0.23
0.9	189.6	0.3278	0.9326	3.26	3.48	—0.22	—0.18
1.0	198.6	0.2699	1.000	0.00	0.00	0.00	0.04

*Calculated from equation 1; $r = 1.000$, $a = 0.557$, $b = 1.500$.
§Example: 5.02—5.33 = —0.31.
†Example: Δ Error = $-\Sigma$ Error/n = 0.39/11 = 0.04; —0.31 + 0.04 = —0.27.

number of clear-cut advantages over cams, with the one exception that the functions must be continuous. Linkages are essentially straight members jointed together. Only a small number of dimensions need to be held closely. The joints make use of standard bearings and backlash can be minimized by end-loading without inducing high friction loads. The links in effect form a solid chain and are not subject to undue acceleration limitations. High accuracy can be achieved by resorting to jig boring and precision grinding, all done on standard machine tools.

Because of the designer's unfamiliarity with linkage techniques, these advantages are often overlooked. The purpose of this discussion is to present a simple and practical approach to linkage layout. Any designer with the ability to do accurate drawing layout and to perform simple, tabular computations should be able to apply these methods. The techniques may seem tedious and unrewarding to the novice but after a little experience is gained the results should become more than gratifying.

Linkages as Computing Elements: The harmonic transformer and the four-bar linkage, shown in *Fig.* 1, are the two bar linkages most commonly used as function generators. The harmonic transformer consists of a crank, connecting link and slider. It may be driven from the crank end when a rotary input and linear output are desired or from the slide end when a linear input and rotary output are required. Two cranks and a connecting link form the four-bar linkage, whose input and output are both rotary. By assignment of correct values to the various parameters, these linkages will mechanize many single-variable functions. The selection of these values, termed linkage layout, is the subject of this presentation.

Methods are discussed, involving drawing technique and tabular computations, for laying out the harmonic transformer and four-bar linkages. Considerations are given to linkages arranged in series, used to mechanize those functions unmanageable with single linkages. Numerical examples are given to further illustrate the various methods. In cases where high accuracy is required, the final adjustment of parameters, by mathematical methods and by the use of correcting devices, is explained. The use of function generator techniques in the design of mechanical computers, for functions of several variables, is given. The mechanical details of linkages, involving joints, sliders of high accuracy and base structures, are described and illustrated.

Linkage Layout: Before considering methods for linkage layout, it will be helpful to establish a few conventions, shown in *Fig.* 2. The most important one is that of "normalizing". Suppose that the motions of a linkage are plotted on Cartesian co-ordinates, with the input horizontal and the output vertical. As an example, if the linkage is a harmonic transformer, the input would be the crank position and the output would be the slide displacement. Normalizing then simply considers these input and output values in terms of fractions of their total ranges. Their plot would occupy a unit square. This relationship between the input and output motion of a linkage, plotted on a unit square in terms of normalized values, is called the "trace" of a linkage. Normalizing allows for a direct comparison of the traces of different linkages regardless of the input and output ranges.

In layout work, the difference between a trace and a diagonal line through its end points is often convenient to use. This difference is termed the "deviation" of a linkage. A reversal in the algebraic sign of the deviation, which is the same as obtaining the deviation from a mirror image of the trace, is equivalent to simply interchanging the input and output scales. This characteristic of either the trace or deviation becomes useful in some linkage layouts. In effect, the deviation may be used to greatly increase the scale of the layout. A comparison between the trace of a linkage and the required trace is referred to the "fit" of the linkage. The actual "error" of the fit is the difference between these traces, expressed as a percentage. Usually it is sufficient to determine the error at eleven equally spaced input intervals (each 10 per cent of input travel), depending upon the shape of the linkage trace. It should be noted that this difference is identical, whether obtained from a comparison of traces or deviation curves. In later numerical examples, these conventions will be further illustrated.

Linkage layout is still largely an art, an almost personal relationship between the designer and his problem. As a general prerequisite to successful design, nothing can replace experience. The designer has a variety of methods available, but will soon develop a preference or style of his own. For this reason, the methods described below should rightly be considered points of view rather than rigid prescriptions.

Our experience has shown the following five methods

to be useful: they are the overlay method[1,3,4], linkages in series[1,4], atlas method[1,4,5], multiple-computation method[1] and the b^2 method. The overlay and linkages in series methods are semigraphical and involve computations only for the purposes of checking the fit exactly and for improving the fit if desirous. They are used when exact fits must be closely approached. Considerable discussion, together with numerical examples of actual linkage layouts, appears later for these two methods. In comparison, the remaining three methods have proven to be less useful and have been employed less frequently. However, these remaining methods will be briefly described below.

As the name might imply, the atlas method makes use of a compilation of linkage characteristics. These appear in sets of plots or tables, containing the input-output data of a variety of linkages. The linkage parameters are varied in a systematic manner, covering a wide selection. These data may be compiled by the designer from his own experience or they may be obtained from published sources (for example, the material in reference 5). By comparison of the linkage requirements, expressed either as a plot or in tabular form, with the atlas, the general region may be selected. Since the parameters vary in sequence, visual interpolation among adjacent values may be used. This method is tedious and generally does not lead to very close fits, particularly if the atlas data is very wide-spread in the region being explored. However, this atlas method is often useful in establishing an initial choice of linkage parameters for use with a more productive method.

Probably the most laborious method is that of multiple-computation. This method consists of selecting a set of linkage parameters, from which the output is calculated. By a systematic variation of parameter and numerous recalculations, a satisfactory fit may be obtained. The use of tabular computing methods and interpolation between selected parameters helps in narrowing-down the process. However, unless some quick method is used to select the initial

[1]References are tabulated on Page 209.

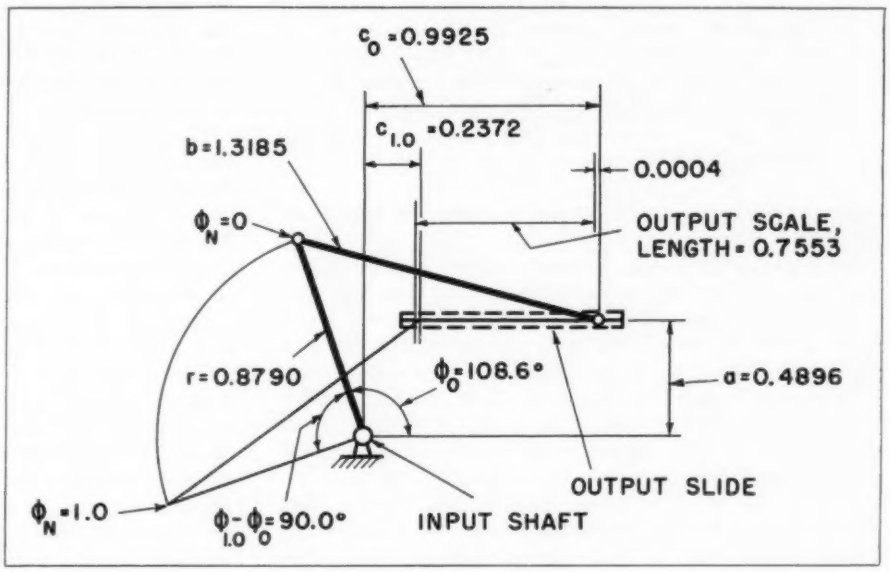


Fig. 5—Right—Dimensions of actual harmonic transformer equal to unit linkage parameters multiplied by ratio 0.7553/0.8593

Fig. 6—Below—Layout of unit four-bar linkage by overlay method

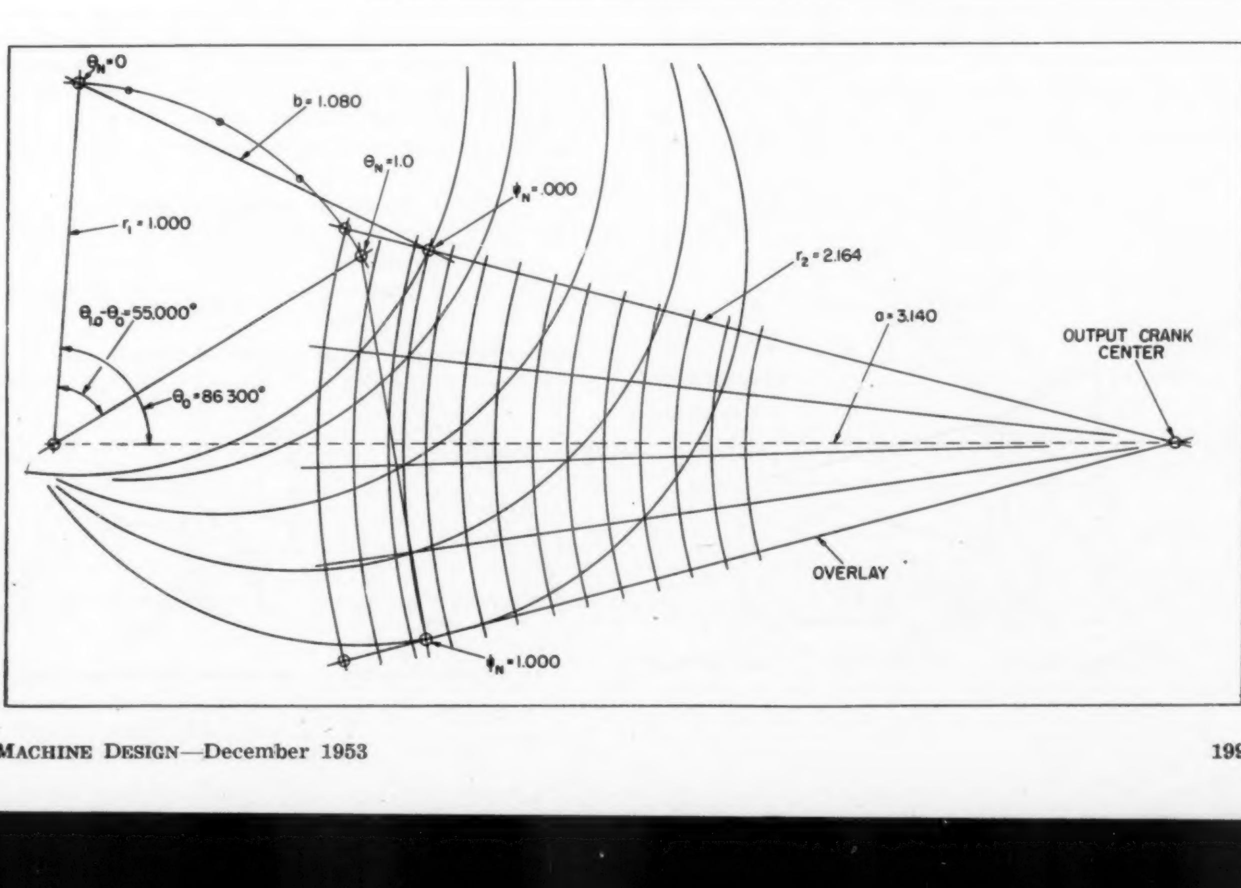


Table 3—Input-Output Requirements for a Four-Bar Linkage

Input*		Output†		Deviation
θ	θ_N	ψ	ψ_N	ε
0	0.0000	−3.8278	0.0000	0.00
20	0.1429	−1.7936	0.2766	13.37
60	0.4286	0.2119	0.5493	12.07
100	0.7143	1.8688	0.7746	6.03
140	1.0000	3.5265	1.0000	0.00

*Required range = 55.000 degrees. †Required range = 28.870 degrees.

Table 4—Calculation of Unit Four-Bar Linkage

θ_N	θ	ψ^*	ψ_N	ε_{mech}	ε_{req}	Error	Zero Set
0.0000	86.300	165.735	0.0000	0.00	0.00	0.00	−0.03
0.1429	78.446	173.687	0.2754	13.25	13.37	0.12	0.09
0.4286	62.727	181.627	0.5504	12.18	12.07	0.11	0.08
0.7143	47.013	188.074	0.7738	5.95	6.03	−0.08	−0.11
1.0000	31.300	194.606	1.0000	0.00	0.00	0.00	−0.03

*Calculated From Equations 4 to 7; $r_1 = 1.0000$, $r_2 = 2.1645$, $a = 3.1398$, $b = 1.0823$.

Table 5—Calculation of Exact Fit of a Four-Bar Linkage

				Error	
ψ_N*	ψ_{req}	ψ_{mech}	$\Delta\psi$	(degrees)	(per cent)
0.0000	0.000	0.000	0.000	−0.005	−0.02
0.2766	7.990†	7.952§	−0.038[1]	−0.043[2]	−0.15
0.5493	15.841	15.892	0.051	0.046	0.16
0.7746	22.347	22.347	−0.008	−0.013	−0.05
1.0000	28.850	28.871	0.021	0.016	0.06

* ψ_N values from Table 3. † Example: $0.2766 \times 28.850 = 7.990$. § Example: $173.687 - 165.735 = 7.952$; 173.687 and 165.735 from Table 4. [1] Example: $7.952 - 7.990 = -0.038$. [2] Example: $-\Sigma(\Delta\psi)^2/n = -.005$; $-0.038 - 0.005 = -0.043$.

parameters, it is practically impossible to arrive at a suitable fit.

The b^2 method too involves direct computation. However, it usually requires less labor on the designer's part and leads towards a more direct solution. In the previous methods discussed, the linkage parameters are adjusted until the required input-output relationship is obtained. This method differs in that a function of one of the linkage parameters is adjusted to fit the given input and output values. In a harmonic transformer, for example, the parameter may be b^2, the square of the link length. The equations for b^2 are particularly simple; hence b^2 is easily calculated. If the adjustment is exact, the calculated values of b^2 will be a constant over the input range. The object then is to keep changing the various parameters until b^2 is made as nearly constant as possible. When this point is reached, the link length is then equal to the square-root of b^2, thus completely defining the linkage. The one disadvantage of this method, in addition to the tedious computations involved, is that the fit is expressed in terms of b^2, or b, and not as a direct comparison of output values. Even after a satisfactory adjustment of b^2 is made, it then became necessary to make this additional error calculation. In both the multiple-computation and b^2 methods, the designer is more or less groping in the dark, hoping by brute force to arrive at satisfactory fits.

Overlay Method: As previously mentioned, our experience has shown the overlay method to be the most productive one for linkage layout. It leads to a more apparent and direct solution, thus avoiding the "groping" present in other methods. To illustrate this method, two numerical examples will be presented. The first example considers a harmonic transformer. The other one involves a four-bar linkage.

The input-output requirements for the first linkage are in TABLE 1. These data give the results of experimental measurements and are typical of the initial information handed to the designer. It is specified that the input be rotary and the output be linear. This requires that the linkage be a harmonic transformer, driven from the crank end of the linkage. The output range is given as 0.7555-inch. Other than being rotary, the input range is arbitrary and is left to the choice of the designer. Quite often, this input range is also specified.

Input and output data are designated by ϕ and c, respectively. The "normalized" values are indicated by the subscript N. "Deviation" in percent, is shown by ε. As seen in TABLE 1, eighteen entries are given for the experimental data. As will be shown later in the discussion, it is desirable to reduce the data to eleven evenly spaced input values. This is accomplished by plotting the deviation, as shown in *Fig.* 3, and then reading-off the desired values, ε', at even 0.1 intervals of ϕ_N. From ε', the required normalized output, c_N', can be determined simply by dividing by 100 and adding to the normalized input, ϕ_N. The use of the deviation rather than the normalized output greatly increases the scale of the plot and serves to smooth-out the data. If the deviation were plotted on the same unit square as the trace and if the vertical scale were so chosen that the maximum point of 11.9 fell near the top line, it would be seen that the scale was effectively increased by approximately eight-times ($100/11.9 \approx 8$). An increase in the actual size of the trace by eight times, would accomplish the same result, except that the plot would become unmanageable due to its size. These

Fig. 7—Dimensions of unit four-bar linkage

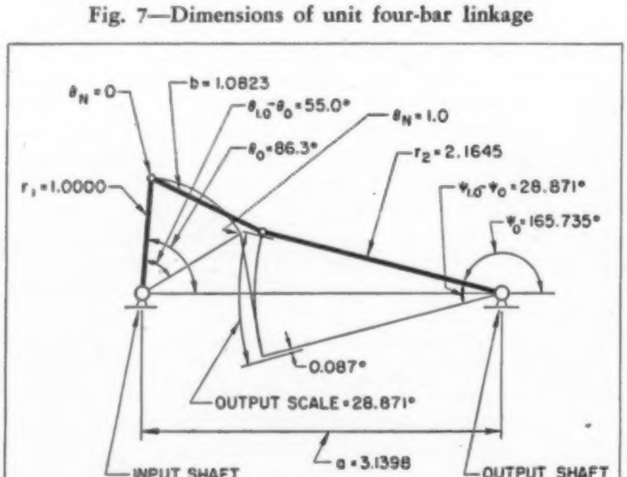
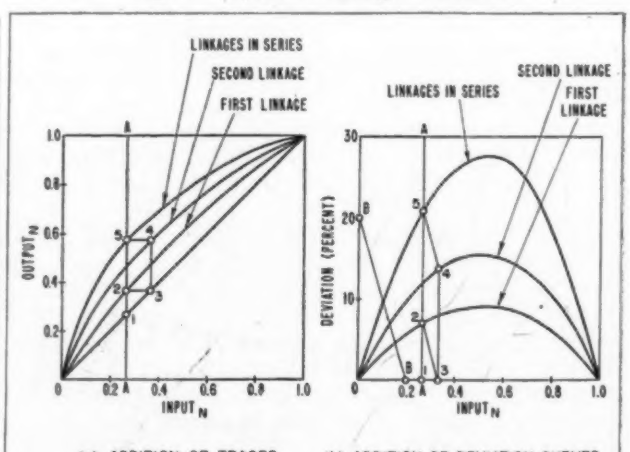


Fig. 8—Layout of linkages in series



smoothed values of the required output, appearing in the last two columns of TABLE 1, are used in laying out the linkage. In addition to obtaining evenly spaced values, this smoothing procedure becomes important, particularly if the experimental data are scattered.

This overlay method is graphical and starts from estimated values of all but two of the linkage parameters, *Fig.* 1, which are related by the expression:

$$c = r \cos \phi + [b^2 - (a - r \sin \phi)^2]^{1/2} \quad \text{..........(1)}$$

where c = output (slide) displacement, r = crank radius, ϕ = input (crank) position, a = vertical height of slide, and b = length of connecting link. For convenience the crank radius is taken as unity. After the ϕ range is chosen, the two parameters to be determined are then a and the initial value of ϕ_0. This linkage is referred to as the "unit linkage". The layout is shown in *Fig.* 4.

The procedure of the overlay method is as follows:

1. Unless otherwise specified, assume a value of the ϕ range. For this illustration where the function is well behaved, i.e., where the deviation curve is smooth and fairly symmetrical, a value of 90 degrees is assumed. Our experience has shown values of 60 to 100 degrees to be workable figures.

2. Lay out a sector with a radius of unity and an included angle of 90 degrees. The actual length of the radius on the layout drawing is taken as 5 inches. From experience again, this has been found to be a convenient scale.

3. Layoff points on the arc according to the normalized input ϕ_N, which in this case is equal to ten equal increments of 9 degrees each.

4. Assume a value of b equal to 1.500 or 7.5 inches in length. The value of b has been found many times to be quite arbitrary and any value between 1.0 to 3.0 is often satisfactory.

5. From each point on the arc of the sector trace an arc of radius equal to b. As seen in *Fig.* 4*a*, the pattern of these traces is symmetrical about the sector. If the values of ϕ_N are evenly spaced, as shown in this example, it is readily seen that only one side of the pattern is required. This symmetry would not exist if the ϕ_N values were unevenly spaced. The entire pattern of the traces then would be needed. This would unnecessarily add to the drawing process.

6. On a separate transparent sheet, lay out the "overlay". As seen in *Fig.* 4*b*, the overlay is simply a series of parallel lines, varying in length, but each proportioned off into ten divisions according to the required normalized output c_N'. The length of the longest line is equal to 10 inches. It is convenient to draw this overlay with the sheet superimposed on a grid system of 10 lines to the inch.

7. Place the overlay over the traces of the radius b, either right-side up or right-side down, and position it so that all eleven points on some one line of the overlay intersect corresponding traces. This position, as shown on *Fig.* 4*c*, then establishes the location of the slide output of the unit linkage. The two remaining parameters are scaled-off as: a = 0.557-inch and ϕ_0 = 108.6 degrees, thus completely defining the linkage.

If a close fit is not obtained for all intersections, in step 7, it is necessary to vary either or both the ϕ range and b and to repeat the layout. Usually several trials are required before a satisfactory fit is obtained. However, as in this illustration where the function

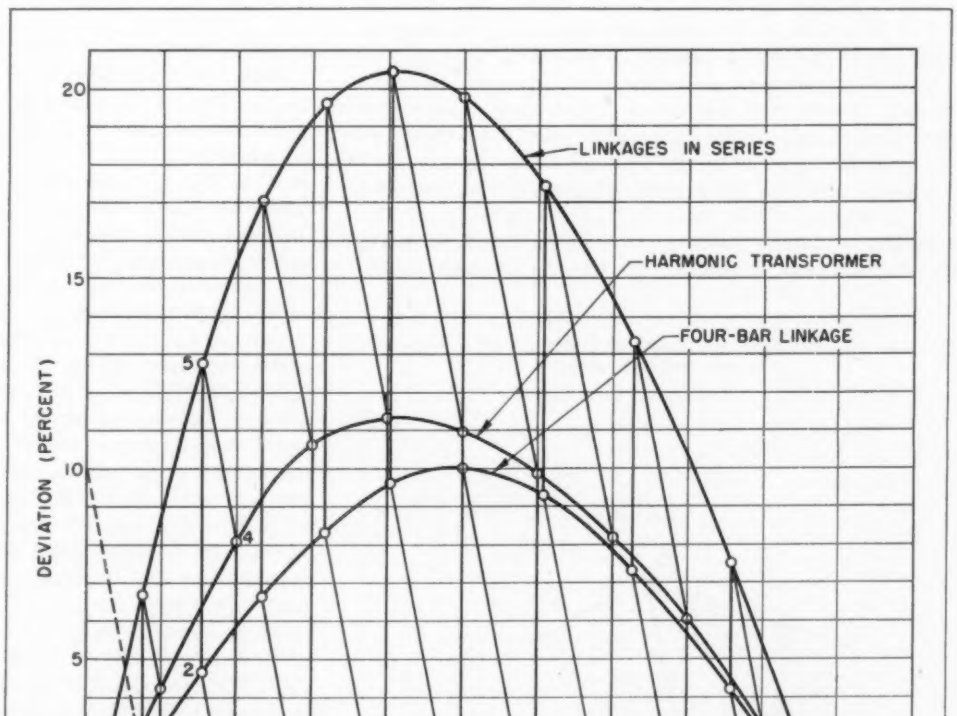


Fig. 9—Layout of a four-bar linkage and a harmonic transformer in series

is well behaved, one or two trials are often sufficient. If a satisfactory fit is still not obtained, it will be necessary to resort to additional procedures. These will be considered later in the article.

From the now determined parameters of the unit linkage, the output displacement is calculated, from Equation 1, in TABLE 2. The positional error is shown by comparing the mechanized deviation, ϵ_{mech} with the required deviation, ϵ'. The maximum positional error may be reduced by applying "zero set" or a least-squares fit. The expression is obtained as follows. Suppose an amount, Δ, is added to each error value, e, then squared and totalled, giving R^2.

$$(e_0 + \Delta)^2 + (e_1 + \Delta)^2 + (e_2 + \Delta)^2 + \ldots (e_n + \Delta)^2 = R^2 \qquad (2)$$

where n is the number of entries in the table, eleven in this illustration. R^2 is made a minimum by setting the differential $d\Delta/dR$ equal to zero, from which the expression

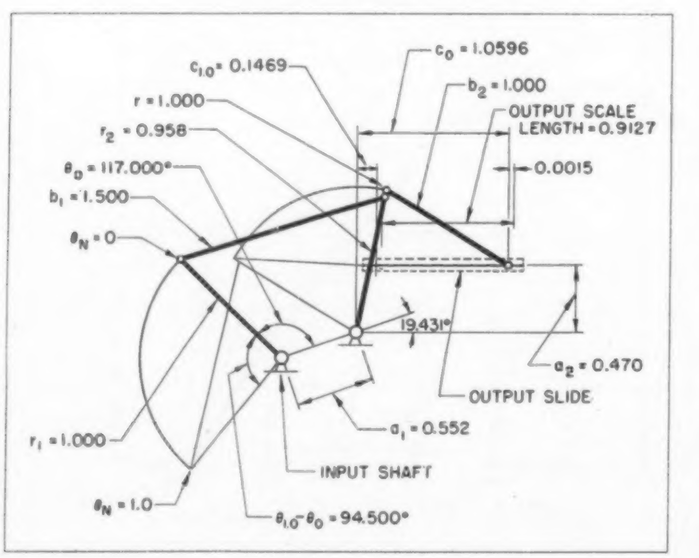
$$\Delta = - \frac{\Sigma e_n}{n} \qquad (3)$$

is obtained, where Σe_n is the algebraic sum of the eleven error figures. In TABLE 2, Δ becomes equal to 0.04 and the maximum error is found to be -0.27 per cent. Zero set, in effect, is simply a shift in the output scale. In this case, the shift is equal to 0.04 per cent of the total output travel or 0.0003-inch, to the left or in the negative direction of c.

The required output range of the harmonic transformer was specified as 0.7553-inch. From TABLE 2, the travel of the unit linkage is equal to $1.1292 - 0.2699 = 0.8593$ units. By multiplying the parameters of the unit linkage by the ratio 0.7553/0.8593, the actual sizes in inches are obtained. They become $r = 0.8790$-inch, $a = 0.4896$-inch and $b = 1.3185$-inch. The actual linkage is shown in *Fig.* 5.

This illustration of a harmonic transformer has considered the linkage as being driven from the crank end. The same overlay procedure could be used for the opposite case, with the linkage being driven from the slide end. The only difference would be that the arc of step 3 would be subdivided according to the normalized output, and the overlay of step 6 would be proportioned according to the normalized input.

Fig. 10—Combination of a four-bar linkage and a harmonic transformer in series



This would result in unevenly spaced values of ϕ_N and would require that the entire pattern of step 5 be drawn. However, the output range is now angular and can not be adjusted to an exact value by simply varying the scale of the linkage. For such a case, it would be much simpler to reverse the algebraic signs of the deviations and to consider the linkage as driven from the crank end. This would allow the output or crank travel to be chosen exactly. The calculated fit of the linkage would of course be in terms of crank position, rather than in terms of slide displacement, as before.

The input-output requirements for the second linkage are shown in TABLE 3. It is specified that both the input and output ranges be rotary, thereby requiring the use of a four-bar linkage. The input range is specified as 55.000 degrees and the output range as 28.865 degrees. Normalized values and the deviation are determined as before. Inasmuch as only five entries are given, it was not necessary to carry-out the "smoothing" procedure. In some cases, however, it might be advisable to interpolate (mathematically) when only a few points are given to obtain the usual eleven points. Only the five points will be considered in this example.

The overlay procedure for the four-bar linkage is very similar to that of the harmonic transformer. The parameters, *Fig.* 1, are related by the expressions:

$$c = (a^2 + r_1^2 - 2\, a r_1 \cos \theta)^{1/2} \qquad (4)$$

$$\alpha = \sin^{-1} \left(\frac{r_1}{c} \sin \theta \right) \qquad (5)$$

$$\beta = \cos^{-1} \frac{c^2 + r_2^2 - b^2}{2\, c\, r_2} \qquad (6)$$

$$\psi = 180 - \alpha - \beta \text{ deg} \qquad (7)$$

where θ = input (crank) displacement, r_1 = input (crank) displacement, r_1 = input crank radius, ψ = output (crank) displacement, r_2 = output crank radius, a = distance between crank centers, b = length of connecting link, and c, α and β are according to *Fig.* 1. For convenience, the input crank radius is taken as unity, designating the linkage as the "unit linkage". With the θ and ψ ranges initially chosen, three parameters are determined by the overlay method. They are r_2, a and the initial value of θ_0. The layout is shown in *Fig.* 6.

Except for the construction of the overlay, the procedure is identical to that for the harmonic transformer. As shown in *Fig.* 6, the overlay now becomes a series of arcs of varying radius, with the included angle equal to the output crank range, and proportioned off according to the required normalized output ψ_N. It is seen that each arc is associated with a particular radius, thereby introducing the third parameter, r_2, which must be determined.

Establish the position of the overlay as before, with the points on any one arc of the overlay intersecting corresponding points on the traces of the end of the connecting link. This position, as shown on *Fig.* 6, then locates the output crank center. Two of the parameters are scaled-off from the layout as: $\theta_0 = 86.300$ degrees and $a = 3.140$ inches. The remaining parameter is measured from the overlay as: $r_2 = 2.164$ inches, thus completely defining the unit linkage. If some particular value of a is required, the linkage may be scaled accordingly.

The output displacement is calculated in TABLE 4, from Equations 4, 5, 6 and 7. Before proceeding with the entire table, it is best to calculate only the first

Table 6—Input-Output Requirements for a Four-Bar Linkage and Harmonic Transformer in Series

Experimental Data: Input* θ	Input* θ_N	Output† c	Output† c_N	Deviation ε	Linkage Data: Input θ_N	Dev. ε'§	Output c_N
1000	0.0000	109.33	0.0000	0.00	0.0	0.00	0.0000
1090	0.0667	103.63	0.1269	6.02	0.1	8.75	0.1875
1170	0.1259	98.875	0.2328	10.69	0.2	15.54	0.3554
1250	0.1852	94.400	0.3324	14.72	0.3	19.25	0.4925
1350	0.2593	89.547	0.4404	18.11	0.4	20.42	0.6042
1450	0.3333	85.440	0.5319	19.86	0.5	19.77	0.6977
1550	0.4074	81.861	0.6118	20.42	0.6	17.62	0.7762
1650	0.4815	78.724	0.6814	19.99	0.7	14.40	0.8440
1750	0.5556	75.961	0.7429	18.73	0.8	10.30	0.9030
1850	0.6296	73.522	0.7972	16.76	0.9	5.45	0.9545
1950	0.7037	71.333	0.8460	14.23	1.0	0.00	1.0000
2050	0.7778	69.346	0.8902	11.24			
2150	0.8519	67.520	0.9308	7.89			
2250	0.9259	65.918	0.9665	4.06			
2350	1.0000	64.414	1.0000	0.00			

*Rotary input required, 54.5 deg. †Linear output required. §Read from *Fig.* 9, linkages in series curve.

Table 7—Calculation of the Unit Four-Bar Linkage for the Combination of a Four-Bar Linkage and Harmonic Transformer in Series

Input-Output Requirement θ_N	ε'*	ψ_N	Output Calculations θ	ψ§	ψ_N	Error	$\Delta\psi$
0.0	0.00	0.0000	117.000	59.069	0.0000	0.00	0.000
0.1	3.10	0.1310	126.450†	68.483	0.1347	0.37	9.414
0.2	5.83	0.2583	135.900	77.291	0.2607	0.24	18.222
0.3	8.00	0.3800	145.350	85.529	0.3786	−0.14	26.460
0.4	9.55	0.4955	154.800	93.223	0.4887	−0.68	34.154
0.5	10.00	0.6000	164.250	100.395	0.5913	−0.87	41.326
0.6	9.37	0.6937	173.700	107.057	0.6866	−0.71	47.988
0.7	7.82	0.7782	183.150	113.226	0.7749	−0.33	54.157
0.8	5.70	0.8570	192.600	118.921	0.8564	−0.06	59.852
0.9	3.05	0.9305	202.050	124.159	0.9313	0.08	65.090
1.0	0.00	1.0000	211.500	128.960	1.0000	0.00	69.891

*Read from *Fig.* 9, four-bar linkage curve. †Example: 0.1 (94.5) + 117.000 = 126.450 deg. §Calculated from Equations 4 to 7; r_1 = 1.000, r_2 = 0.958, a = 0.552, b = 1.500.

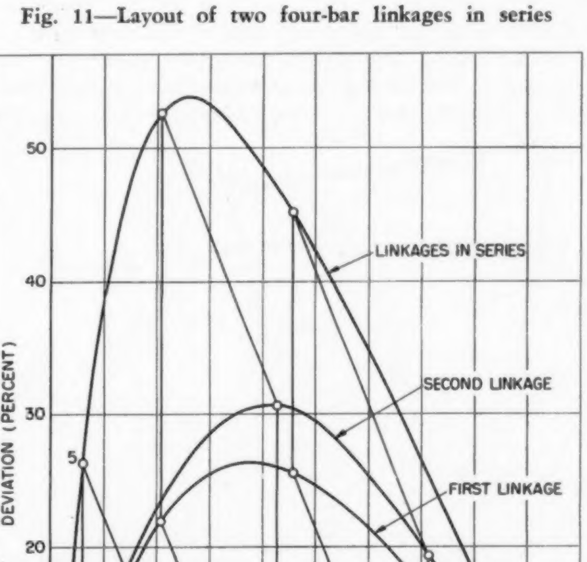
and last lines in order to determine the output range. It usually will be found that this range differs a small amount from that required. To arrive at a more exact value, it is necessary to vary the parameters slightly and to repeat the calculations until the correct value is obtained. Three or four such trial-and-error computations usually are sufficient. From TABLE 4, it is seen that the output range is equal to 194.606 − 165.735 = 28.871 degrees, compared to the required value of 28.870 degrees, with r_1 = 1.0000 inches, r_2 = 2.1645 inches, a = 3.1398 inches and b = 1.0823 inches. The actual linkage is shown in *Fig.* 7.

Suppose, for example, that the specified range had been 28.850 degrees, rather than 28.870 degrees. Now, because the output range of the linkage is not equal to this required value, the usual method of comparing the deviations will no longer give the exact fit. This fact is seen from the definition of the convention "deviation", explained earlier. Inasmuch as the displacement is angular or nondimensional, it is not possible to adjust the range by assigning a scale or size to the linkage, as was true with the harmonic transformer. It then becomes necessary to determine the fit by comparing the actual displacements, in degrees, as shown in TABLE 5. The required output displacement, ψ_{req}, is obtained by multiplying the 28.850-degree range by the normalized values, ψ_N, obtained from TABLE 4. The mechanized displacement, ψ_{mech}, is obtained from TABLE 4, as the difference of successive values subtracted from the first ψ value. Taking this difference and applying zero-set give the error values in degrees. In the last column, these values are converted to percentages of the required range. It is seen that these exact errors are larger than those given by a comparison of deviation values. The sum of the squared errors is 546, compared to 284. In this illustration, the actual difference in error values given by either method is small and perhaps negligible. But if an exact check of the fit is required, this last method should be used, especially if the difference between the required and mechanized ranges is large compared to the error values. Because of this adjustment of the output range, together with the added calculation of the exact fit, the layout of a four-bar linkage is always more tedious than that for the harmonic transformer.

These examples show fitting errors on the order of 0.3 per cent or less and are typical values that may be expected from the overlay method. Only the end results are given, and are in no way a measure of the labor involved. For well-behaved functions, only one or two trials are usually necessary in laying out the linkage. In cases where the functions are not so well behaved, the designer should be prepared to make several trials. However, unless good drawing technique is practiced, together with lots of patience and persistence, this kind of results can not be achieved.

Linkages in Series: In many cases where the deviation is large and unsymmetrical, it will be found that a single linkage will not be satisfactory. It then becomes necessary to use linkages in tandem or series, with the output of one linkage driving the input of the next linkage. Usually, two linkages in series are sufficient. For example, a four-bar linkage may be used in series with a harmonic transformer or with another four-bar linkage. Or two harmonic

Fig. 11—Layout of two four-bar linkages in series



transformers may be linked together, back-to-back, by making the slide common to both linkages.

The layout procedure for linkages in series involves the selection of two traces, which when combined or added together give the required trace. The procedure for adding traces is relatively simple and is shown in *Fig. 8a*. First, the two traces and the diagonal are plotted on the same unit square. Draw a vertical line *A-A*, intersecting the diagonal at 1 and the trace of the first linkage at 2. The length 1-2 is the deviation of the first trace. The height of 2 is the output of the first linkage, which when projected horizontally to 3 on the diagonal, becomes the input to the second linkage. Point 3 is projected vertically to 4, the output of the second linkage. Point 4 is projected horizontally to line *A-A*, giving 5, the output of the second linkage corresponding to the input of the first linkage, which is one point on the trace of the combined linkages. The lengths 3-4 and 2-5 are both equal to the deviation of the second trace, at an input value equal to the output of the first linkage. In other words, 5 may be considered as simply being located by adding this second deviation 2-5, to the first deviation 1-2.

A similar construction, using deviation values directly, is shown in *Fig. 8b*. The points are numbered the same as in *Fig. 8a* and have the same significance. Lines 2-3 and 4-5 are both parallel to a line *B-B*, drawn through equal numerical values on the deviation and input scales. The numerical length 1-3 is equal to the deviation 1-2, which added to the input, point 1, gives the output of the first linkage, point 3, or the input of the second linkage. Adding the two deviations as before gives point 5 on the deviation curve of the combined linkages. Again it is seen that by using deviation values the scale of the layout is considerably increased. For example, if the maximum deviation were approximately 20 per cent and both constructions made in the same size plots, the effective scale and accuracy of the deviation method would be five times that of the first method.

In an actual layout, the deviation curve for the linkages in series is given. A curve for the first linkage is selected, usually lying midway below, and the curve of the second linkage constructed. The points are determined as before, but in the order of 3, 2, 5 and 4. Occasionally the designer may start with a known linkage, from some previous problem, and plot its deviation as the first linkage. This eliminates the fitting error and thus specifies the second linkage more closely. After a little experience, it is rare to choose a first linkage which leads to an unmanageable second linkage. Two numerical examples are given to illustrate this procedure. The first one shows a four-bar linkage driving a harmonic transformer. The other one deals with two four-bar linkages in series.

The input-output requirements for a four-bar linkage and harmonic transformer in series are given in

Table 8—Calculation of the Unit Harmonic Transformer for the Combination of a Four-Bar Linkage and Harmonic Transformer in Series

θ_N	ϕ*	c†	c_N	ε_{mech}	ε'§	Error	Zero Set
0.0	78.500	1.0596	0.0000	0.00	0.00	0.00	−0.16
0.1	87.914	0.8848	0.1915	9.15	8.75	0.40	0.24
0.2	96.722	0.7352	0.3554	15.54	15.54	0.00	−0.16
0.3	104.960	0.6101	0.4925	19.25	19.25	0.00	−0.16
0.4	112.654	0.5064	0.6061	20.61	20.42	0.19	0.03
0.5	119.826	0.4202	0.7005	20.05	19.77	0.28	0.12
0.6	126.488	0.3479	0.7797	17.97	17.62	0.35	0.19
0.7	132.657	0.2865	0.8470	14.70	14.40	0.30	0.14
0.8	138.352	0.2337	0.9049	10.49	10.30	0.19	0.03
0.9	143.590	0.1876	0.9554	5.54	5.45	0.09	−0.07
1.0	148.391	0.1469	1.0000	0.00	0.00	0.00	−0.16

* $78.500 + \Delta\psi$, from Table 7. † Calculated from Equation 1; $r = 1.000$, $a = 0.470$, $b = 1.000$. § ε' From Table 6.

Table 9—Input-Output Requirements for Two Four-Bar Linkages in Series

Input*		Output†		Deviation
θ	θ_N	ϕ	ϕ_N	ε
−20	0.0000	−0.3567	0.0000	0.00
−10	0.1429	−0.0470	0.6074	46.45
10	0.4286	0.1004	0.8964	46.78
30	0.7143	0.1371	0.9684	25.41
50	1.0000	0.1532	1.0000	0.00

*Rotary input required, 55.000 deg. †Rotary output required, 43.927 deg.

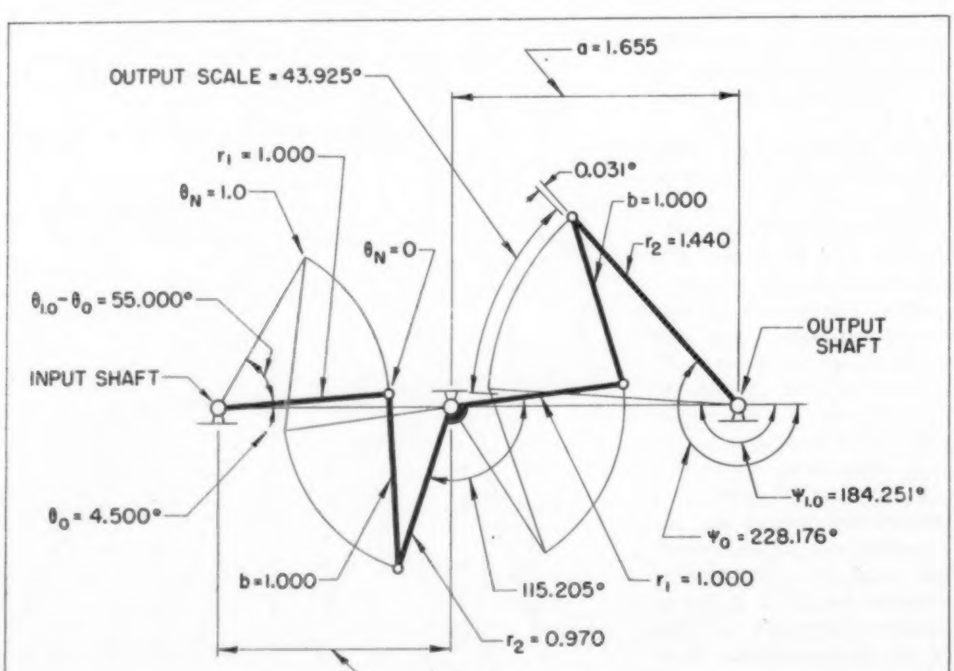


Fig. 12 — Two four-bar linkages in series

Table 10—Calculation of the First Unit Linkage of Two Four-Bar Linkages in Series

θ_N	θ_1	ψ_1†	ψ_{1N}	$\Delta\psi_1$	ε_1
0.0000	4.500	251.605	0.0000	0.000	0.00
0.1429	12.354*	231.493	0.3151	20.112	17.22
0.4286	28.073	207.598	0.6895	44.007	26.09
0.7143	43.787	195.542	0.8784	56.063	16.41
1.0000	59.500	187.781	1.0000	63.824	0.00

*Example: 0.1429 (55.000) + 4.500 = 12.354 deg. †Calculated from Equations 4 to 7; $r_1 = 1.000$, $r_2 = 0.970$, $a = 1.350$, $b = 1.000$.
Subscript 1 refers to first linkage in series.

Table 11—Calculation of the Second Unit Linkage of Two Four-Bar Linkages in Series

θ_N	θ_2*	ψ_2†	ψ_{2N}	ε_{mech}	ε_{req}	Error	Zero Set
0.0000	−6.810	228.176	0.0000	0.00	0.00	0.00	−0.07
0.1429	13.302	201.189	0.6144	47.15	46.45	0.70	0.63
0.4286	37.197	188.823	0.8959	46.73	46.78	−0.05	−0.12
0.7143	49.253	185.781	0.9652	25.09	25.41	−0.32	−0.39
1.0000	57.014	184.251	1.0000	0.00	0.00	0.90	−0.07

*−6.810 + $\Delta\psi$, from Table 10. †Calculated from Equations 4 to 7; $r_1 = 1.000$, $r_2 = 1.440$, $a = 1.655$, $b = 1.000$.
Subscript 2 refers to second linkage in series.

TABLE 6. The deviation is plotted in *Fig.* 9 and the usual eleven values read off. A deviation curve is chosen for the first linkage, having a maximum value of 10 per cent, and the second curve is constructed. A visual inspection insures that the two curves are well behaved and should be easily mechanized. By the overlay method, the parameters of the four-bar linkage are determined as: $r_1 = 1.000$ inch, $r_2 = 0.958$ inch, $a = 0.552$-inch and $b = 1.500$ inches. The output calculations are given in TABLE 7. The maximum fitting error is approximately 0.9 per cent and is adequate for the first linkage. In laying out the second linkage, it is not necessary to read off deviation values from *Fig.* 9. The input sector may be laid out directly from the angular output displacement, $\Delta\psi$, given in TABLE 7. The overlay is proportioned off according to the required deviation, ε', of TABLE 6. The harmonic transformer parameters are found to be $r = 1.000$ inch, $a = 0.470$-inch and $b = 1.000$ inch. Output values are calculated in TABLE 8, giving a maximum error, after applying zero-set, of 0.24 per cent. Shown in *Fig.* 10 are the combined linkages, with the output crank of the four-bar linkage and the input crank of the harmonic transformer being common to both linkages. If so desired, the two cranks could be separated by rotating the entire four-bar linkage about its output crank center, the two cranks being separated by an amount equal to the rotation. Also, the distance between the two crank centers and the length of the output scale can be separately adjusted by varying the size of each linkage.

A second illustration, considering two four-bar linkages in series, is also presented. The procedure is exactly as before, except that the first linkage is calculated, from a preliminary overlay construction, and the actual deviation values plotted. In this example, where the deviation values are large, it is very desirable to use calculated values for the first linkage, so that the results of the second layout can be more accurately predicted. Otherwise, the layout might have to be repeated several times, until a satisfactory linkage combination is obtained.

The input-output requirements are given in TABLE 9, with both the input and output ranges being specified. Output calculations for the first linkage are contained in TABLE 10, with the parameters $r_1 = 1.00$ inch, $r_2 = 0.970$-inch, $a = 1.350$ inches and $b = 1.000$ inch. The layout is shown in *Fig.* 11, and the computations of the second linkage are given in TABLE 11 where the parameters are $r_1 = 1.000$ inch, $r_2 = 1.440$ inches, $a = 1.655$ inches and $b = 1.000$ inch. The combined linkages, with the three crank centers arbitrarily shown on a straight line, are con-

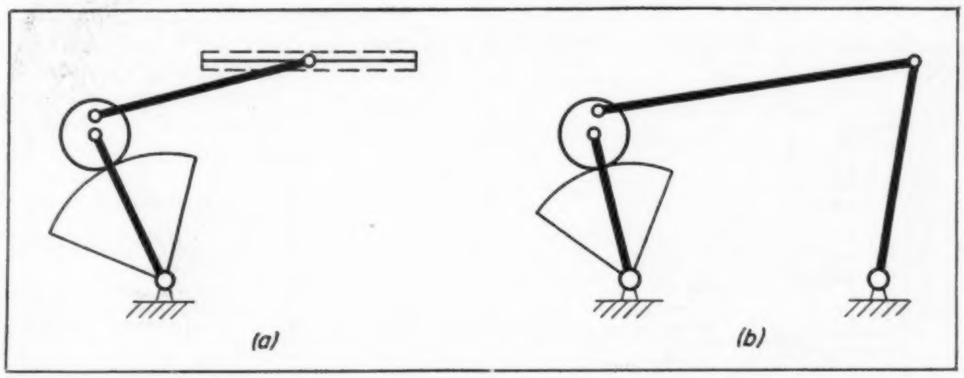
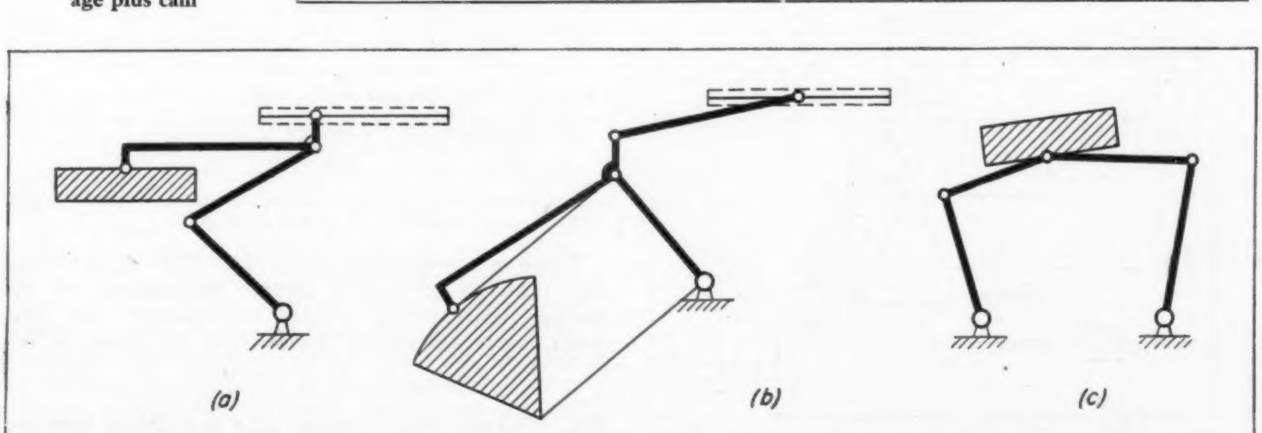


Fig. 13—Right—Eccentric devices used to correct the output of function generators: *a*, harmonic transformer plus eccentric; *b*, four-bar linkage plus eccentric

Fig. 14—Below—Cam devices used to correct the output of function generators: *a*, harmonic transformer plus straight cam; *b*, harmonic transformer plus circular cam; *c*, "broken" four-bar linkage plus cam



tained in *Fig.* 12. It should be noted that, in order to satisfy the directions of the crank rotations, it was necessary to arrange the second linkage as a mirror image from the usual convention, with the crank displacements measured in a clockwise, rather than a counterclockwise, direction. Again, the crank centers may be rotated and the sizes of each linkage adjusted as required. This linkage combination, which mechanizes a very unsymmetrical and sharply-peaked trace, is particularly interesting and is typical of some of the unusual designs encountered.

Final Linkage Adjustment: Many times it is necessary to select a linkage where great precision is necessary, perhaps beyond the scope of the usual methods as previously described. It then becomes necessary to apply mathematical refinements or to employ correcting devices. These are discussed briefly below.

An expression for the harmonic transformer, driven from the crank end, giving the output change resulting from small variations in the linkage parameters is as follows:

$$\Delta c = \Delta \phi \frac{\partial c}{\partial \phi} + \Delta r \frac{\partial c}{\partial r} + \Delta a \frac{\partial c}{\partial a} + \Delta b \frac{\partial c}{\partial b} \quad (8)$$

where each term represents the effect of changing one parameter. The net effect is simply the sum of these terms. By differentiating Equation 1 and collecting terms, the partial derivatives become:

$$\frac{\partial c}{\partial \phi} = -\frac{cr \sin \phi - ar \cos \phi}{c - r \cos \phi} \quad (9)$$

$$\frac{\partial c}{\partial r} = \frac{c \cos \phi + a \sin \phi - r}{c - r \cos \phi} \quad (10)$$

$$\frac{\partial c}{\partial a} = -\frac{a - r \sin \phi}{c - r \cos \phi} \quad (11)$$

$$\frac{\partial c}{\partial b} = \frac{b}{c - r \cos \phi} \quad (12)$$

Considering Δc to be the fitting error, it is seen that by a correct adjustment of the Δ terms, the fitting error might be reduced. The procedure is to first calculate and plot the partial derivatives for the usual eleven values of the normalized input. Next plot the error values. Through a trial-and-error process, find a curve of one of the partial derivatives, or of several summed together, whose shape is generally like that of the error plot. Then assign numerical values to the Δ variations so that the sum of the terms on the right side of Equation 8 is equal to the negative value of the error Δc. If this condition is obtained for all eleven input points, then the fitting error is eliminated by modifying the linkage parameters according to the Δ increments. It is rarely possible to eliminate the errors completely, but this procedure should result in a substantial reduction.

In a similar manner, the errors may be reduced for the harmonic transformer, driven from the slide end, and for the four-bar linkage. The expressions are given below.

For the harmonic transformer:

$$\Delta \phi = \Delta r \frac{\partial \phi}{\partial r} + \Delta a \frac{\partial \phi}{\partial a} + \Delta b \frac{\partial \phi}{\partial b} + \Delta c \frac{\partial \phi}{\partial c} \quad (13)$$

where

$$\frac{\partial \phi}{\partial r} = \frac{c \cos \phi + a \sin \phi - r}{cr \sin \phi - ar \cos \phi} \quad (14)$$

$$\frac{\partial \phi}{\partial a} = -\frac{a - r \sin \phi}{cr \sin \phi - ar \cos \phi} \quad (15)$$

$$\frac{\partial \phi}{\partial b} = \frac{b}{cr \sin \phi - ar \cos \phi} \quad (16)$$

$$\frac{\partial \phi}{\partial c} = -\frac{c - r \cos \phi}{cr \sin \phi - ar \cos \phi} \quad (17)$$

For the four bar linkage:

$$\Delta \psi = \Delta e \frac{\partial \psi}{\partial \theta} + \Delta r_1 \frac{\partial \psi}{\partial r_1} + \Delta r_2 \frac{\partial \psi}{\partial r_2} + \Delta a \frac{\partial \psi}{\partial a} + \Delta b \frac{\partial \psi}{\partial b} \quad (18)$$

where

$$\frac{\partial \psi}{\partial \theta} = \frac{r_1[a \sin \theta + r_2 \sin (\theta - \psi)]}{r_2[a \sin \psi + r_1 \sin (\theta - \psi)]} \quad (19)$$

$$\frac{\partial \psi}{\partial r_1} = \frac{r_1 - a \cos \theta - r_2 \cos (\theta - \psi)}{r_2[a \sin \psi + r_1 \sin (\theta - \psi)]} \quad (20)$$

$$\frac{\partial \psi}{\partial r_2} = \frac{r_2 + a \cos \psi - r_1 \cos (\theta - \psi)}{r_2[a \sin \psi + r_1 \sin (\theta - \psi)]} \quad (21)$$

$$\frac{\partial \psi}{\partial a} = \frac{a - r_1 \cos \theta + r_2 \cos \psi}{r_2[a \sin \psi + r_1 \sin (\theta - \psi)]} \quad (22)$$

$$\frac{\partial \psi}{\partial b} = -\frac{b}{r_2[a \sin \psi + r_1 \sin (\theta - \psi)]} \quad (23)$$

There is always some point (the point of diminishing returns) at which further adjustment of the linkage parameters is not worthwhile. At this point, some correcting device may be added to re-

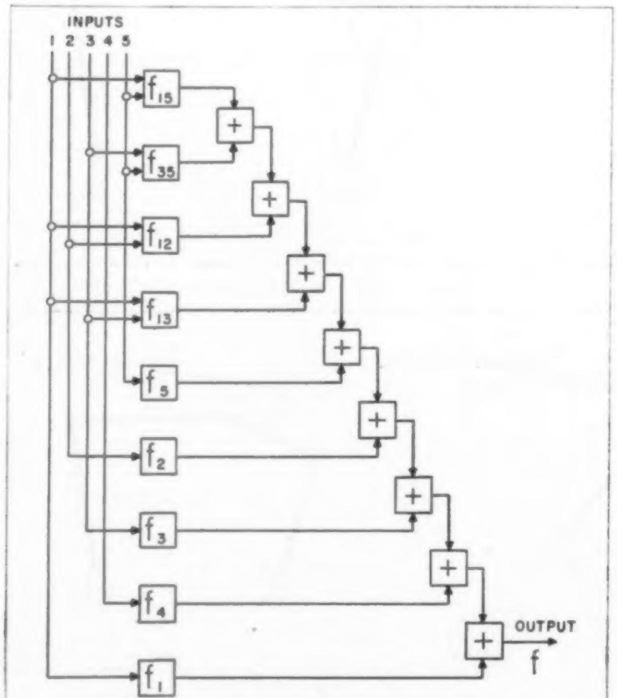


Fig. 15—Left—Block diagram of a mechanical computer using function generator techniques

move the remaining error. Several arrangements, using an eccentric[4] or a cam-and-bell crank combination, are shown. In *Fig.* 13, an eccentric is located between the input crank and the connecting link, either in a harmonic transformer or a four-bar linkage, and in effect changes the length of the connecting link as the crank is rotated. In *Fig.* 14, a cam-actuated bell crank is added to a harmonic transformer. The cam reference, to which the adjustments are added, is straight when the bell crank is located between the link and the slide, and circular when added between the crank and link. In either case, the slide must be raised vertically from its original position by an amount equal to the length of the shorter arm of the bell crank. For the four-bar linkage, the correction may be introduced by "breaking" the link near its center and guiding the point by a cam. Since the required corrections are usually small, compared to the total throw of the linkage, the cam accuracy is low compared to the linkage accuracy. This is particularly true in cases where space permits the use of high-ratio bell cranks on the order of ten to twenty.

No fixed rule can be set as to when to use these correcting devices. Within practical limits, they should be avoided and used only when absolutely necessary. Factors to be considered are the fit required, the shape of the trace being mechanized, the number of units to be built, etc. Associated with this is the problem of predicting, in advance, the difficulty which may be encountered in laying out the linkage. It is not clear what specific characteristics make a function difficult to mechanize. Generally, it is easier to handle a function having a smooth and fairly symmetrical deviation curve, compared to one having steep slopes and irregular inflection points. An experienced designer will often know by inspecting the deviation curve whether or not to expect the need of correcting devices.

Mechanical Computers: In reference 2, mathematical methods are presented, making it possible to use function generator techniques in the design of mechanical computers, for arbitrary functions of as many as five independent variables. Briefly, the mathematics involve the expansion of tabular data, of several independent variables, into a sum of one-function variables plus the products of functions, taking two variables at a time. The expression is:

$$f = f_1 + f_2 + f_3 + f_4 + f_5 + f_{12} + f_{13} + f_{14}$$
$$f_{15} + f_{23} + \ldots f_{45} \quad \ldots\ldots (24)$$

where f_1, f_2, etc., are the single-variable functions and f_{12}, f_{13}, etc., are the products of functions. Function generators may be used to mechanize the single-function terms. The product-of-function expressions may be represented by two function generators driving linkage or slide multipliers. The summing is then performed by a chain of adders. A block diagram of a computer, having five variables or inputs, is shown in *Fig.* 15. It should be noted that the functions are summed in order of increasing size. The output of the largest function, f_1, feeds into the last adder, whose output is the total function f. The next largest function, f_4, drives the next to the last adder, etc., through the chain to the smallest function, f_{15}. This arrangement minimizes the effect of errors resulting from dimensional tolerances in the adder chain.

Linkage adders, *Fig.* 16, whose inputs and outputs are linear, are used in the adder chain. The inexact adder may be used at the low-tolerance end of the chain and the exact adder near the output end of the chain. Both adders consist of three slides, a double crank (two arms 180 degrees apart) and two connecting links. As used in the mechanical computer, the two input slides are common to either the output slide of a function generator or the output slide of a multiplier. The slide displacements, x, referenced from some initial position are related by the expression:

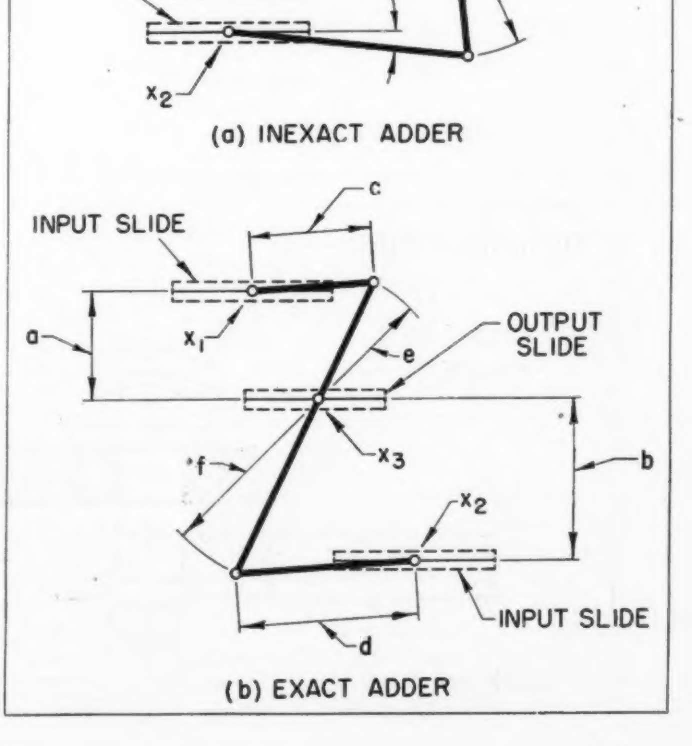
$$x_3 = \frac{b\,x_1 + a\,x_2}{a + b} \quad \ldots\ldots (25)$$

were: $c/d = e/f = a/b$ for the exact adder. This expression is exact for the exact adder and is closely approximated by the inexact adder if the change in the slopes, α and β, of the links are small over the input ranges.

Mechanical Construction: Linkage mechanisms consist of three basic elements: cranks, links and slides. Excepting for the accuracy requirements, the mechanical details of these elements may be similar for all linkages. Three of the most important details, involving joints, sliders and base structures are described and illustrated.

Shown in *Fig.* 17 is a rotary joint using miniature

Fig. 16—Linkage adders for use in summing the function generator outputs of a mechanical computer



pivot bearings. This type of a joint is used to connect a link to a crank or slide, for example in the harmonic transformer. Two bearings are used in each joint. The link is made in two halves, which are doweled and clamped together, forming a clevis at each end. Each half of the clevis carries a bearing in a counterbored recess. A hardened-steel, conical-ended pivot pin is carried in the crank or carriage of the slider. Each half of the link is clamped around the pivot. In the clamped position, the legs of the clevis are slightly spring loaded, thus removing backlash from the joint.

A joint using miniature radial bearings is illustrated in *Fig.* 18. This type of joint is used to connect the crank and output slide of the linkage adder, for example. Two bearings are used in each joint. The outer races of the bearings are carried in the crank. They are separated by a spacer and are retained in the crank by spinning over the edges of the hole. A

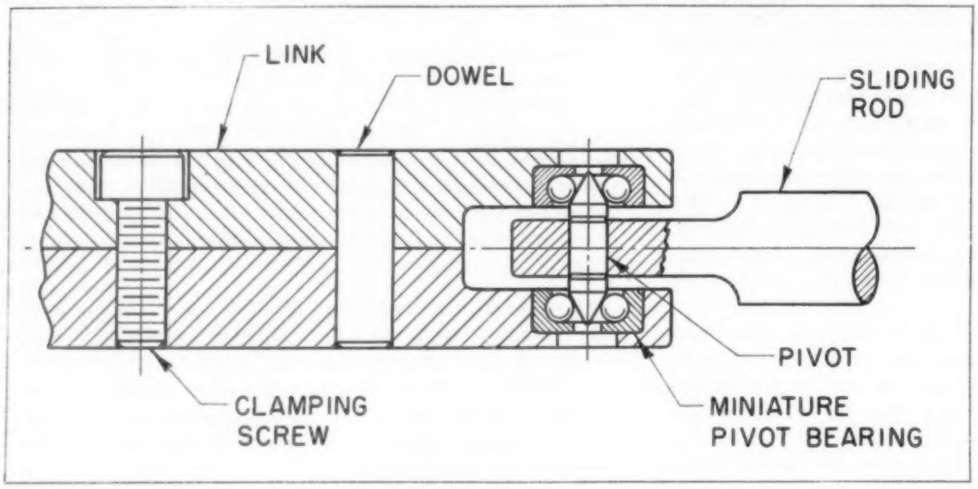


Fig. 17 — Pivot bearing joint

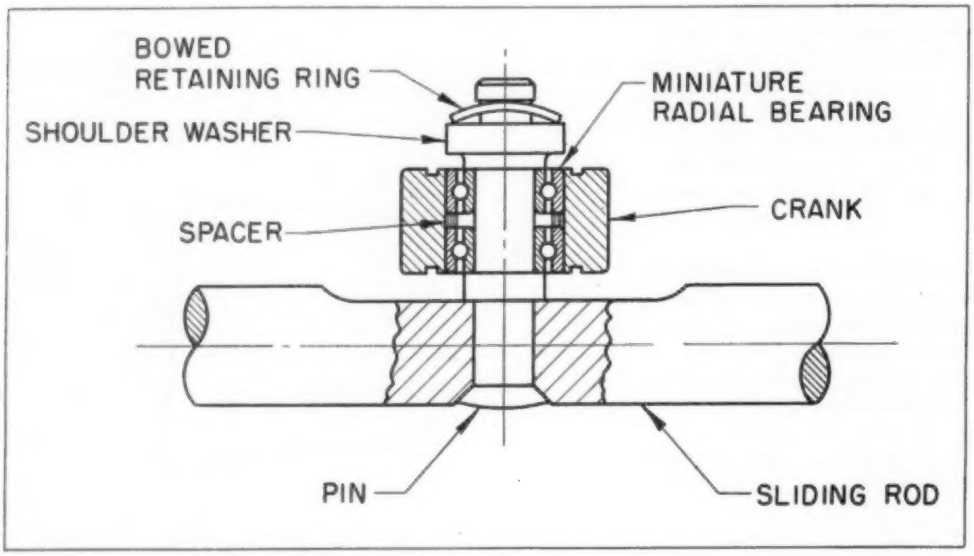


Fig. 18 — Radial bearing joint

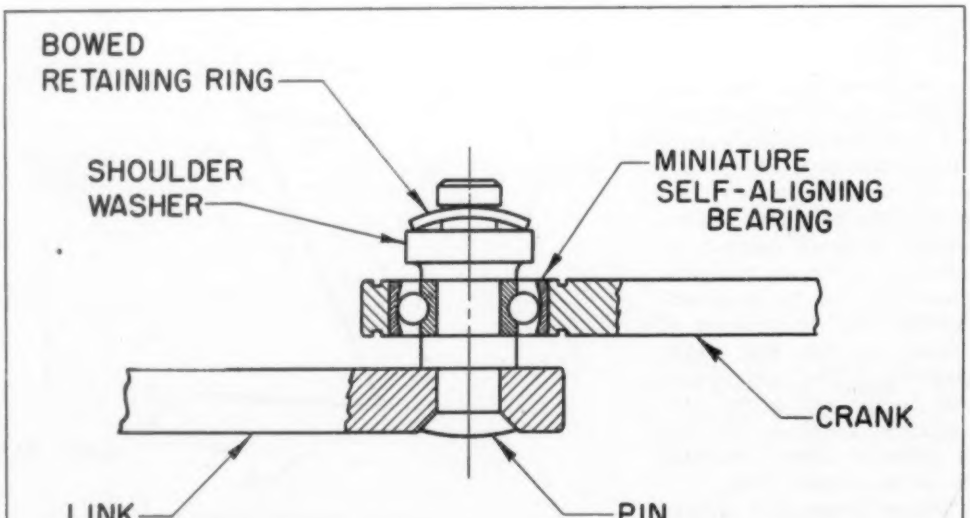


Fig. 19 — Self aligning joint

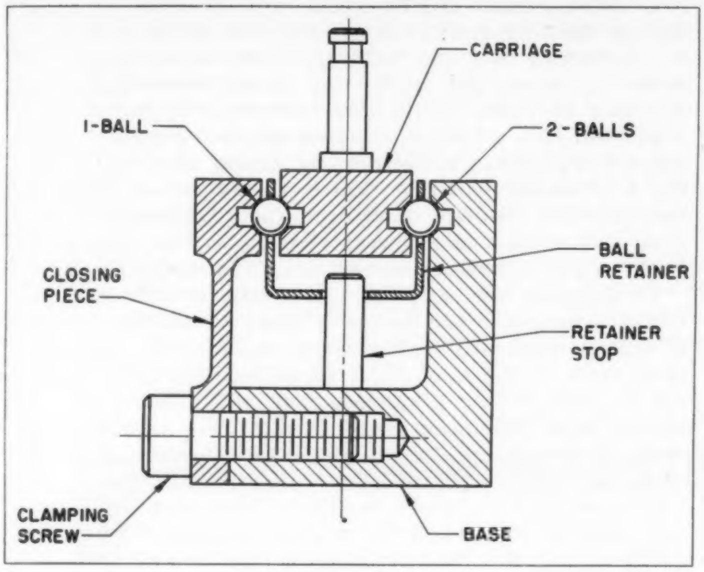


Fig. 20—Three-ball slide

pin fastened to the slider carries the inner races of the bearings. Backlash is minimized by end loading the inner races with a bowed retaining ring, acting through a shouldered washer.

A self-aligning joint, using a miniature self-aligning bearing, is shown in *Fig.* 19. As an example, this type of joint is used in the connection of a link to an adder crank. It permits misalignment between the adder sliders without binding any of the joints. This joint is identical with the radial-bearing type except that only a single bearing is used. Because of the self-aligning features, it is not possible to reduce backlash by end-loading this bearing. This can be accomplished only by the preselection of bearings.

Standard ball bushings form the slides. Two bushings are carried by a channel-shaped bracket. A hardened and ground steel rod is used as the sliding member. Backlash is always present in this type of slide and can be controlled only by the fit of the sliding rod in the bushing. When a minimum-backlash slide is required, the type as shown in *Fig.* 20 should be used. This slider is supported by three balls, hence the name "three-ball" slide. The base of the hardened-steel slide is L-shaped and has a ground V-groove in one leg. A closing piece carrying another parallel groove is doweled and clamped to the base. The web portion of the closing piece is narrowed down and cut away in the center leaving two supporting legs. The carriage contains two parallel V-grooves and is supported on three steel balls. One ball is located on the closing-piece side of the carriage; the other two on the opposite side. In the clamped position the legs of the closing piece are slightly spring loaded, thereby removing all backlash. A ball retainer, together with a stop, is provided to correctly space the three balls. Depending upon the type of connection made to it, the carriage is either drilled to receive a pin or counterbored to carry a pivot bearing. This slide is used only when the backlash or tracking accuracy of the ball-bushing type is not acceptable.

Linkages are mounted on base plates, ground for flatness. Input shafts, carrying the cranks of a four-bar linkage for example, are supported on radial ball bearings. To minimize backlash, two bearings are used with each shaft and preloaded through the inner races by a bowed retaining ring. The bearings are carried in a separate bearing housing which is doweled and bolted or screwed to the base plate. In a mechanical computer these base plates are arranged into a clock-plate structure. Quite often the lower plate may be a rigid surface-plate, with the succeeding plates being attached to it through posts and spacer bushings. Usually it is possible to mount several linkages and adders on one plate. Inputs are introduced by shafts extending through all plates. In order to transmit the output motions from one plate-level to another, the adders may be skewed with the slides being located on separate plates, or identical harmonic transformers may be used at different levels and their motions transferred through a common shaft. The size and compactness again rest upon the ingenuity of the designer.

The foregoing mechanical details involve the support of individual linkages and give methods for minimizing backlash. The assignment of tolerances, to the linear and angular dimensions of the linkage, is a complete subject in itself and is beyond the scope of this article. In reference 6 a complete discussion of this subject is presented.

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Designing Cams with Aid of Computers

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IN cam design, the analysis of the machine motion or timing diagram yields the basic dimensions of the necessary cam profiles and linkage. However, a wealth of information must be supplied to manufacturing personnel in order to make a precision cam that will produce accurate machine motions with reasonably sound dynamic principles. This step, from the basic dimensions to the physical cam, generally embodies extensive and wearisome calculation, especially if tooling is required in the form of master cams. This discussion is concerned with a means of reducing the calculation drudgery by equipment that may be available but not in use. Although IBM accounting equipment is discussed, other types of accounting machines can serve the purpose if they can handle accounting problems.

Cams are designed in many shapes and forms to fit machine conditions and designers' needs. There are plate cams, barrel cams, face cams, end cams, and others. Each operates a follower, such as a shoe or roller, which may be spring returned or positively driven in each direction. In each type, the follower motion may require specification at all points or it may be sufficient to specify only certain points, following any path between these points. The latter type

generally must give a certain rise or lift in a specified time. The motion during rise may be constant velocity, constant acceleration, or any of the numerous schemes employed to modify these factors when dynamics must be considered.

To produce the master cam or a single cam, a table of cam radii with corresponding cam angles must be supplied. The cam is then cut on a milling machine or other suitable machine tool using point by point settings. The result is a surface with a series of ridges which must be filed down to a smooth profile. The cam radius, cutter radius, and frequency of machine setting determine the extent of filing and final accuracy of the profile. For accurate master cams, settings must often be in half-degree increments calculated to seconds. Preparation of this table may require the solution of six or eight equations for each of these machine settings.

In the author's company, cam and other engineering calculations are of an intermittent nature which only justify the use of a desk calculator. For a number of years, cams were laboriously "punched" out on the calculator with constant reference into and out of trigonometric and other prepared tables. The results were often replete with errors and the time involved was a serious bottleneck. Although engineering calculations could not justify more than desk calculators, the company's accounting department did have an assortment of IBM machines. Lack of knowledge of the equipment's full potentialities on the part of engineering and the inability of accounting personnel to understand engineering terms discouraged attempts to profit by its use. However, the recent acquisition of a newer, electronic-type, computer spurred engineering interest to the extent of attending a two-week part-time course on the use of the machine. The resultant unfolding of potentialities has since shifted all cam and gear calculations and other work such as statistical correlation to IBM equipment.

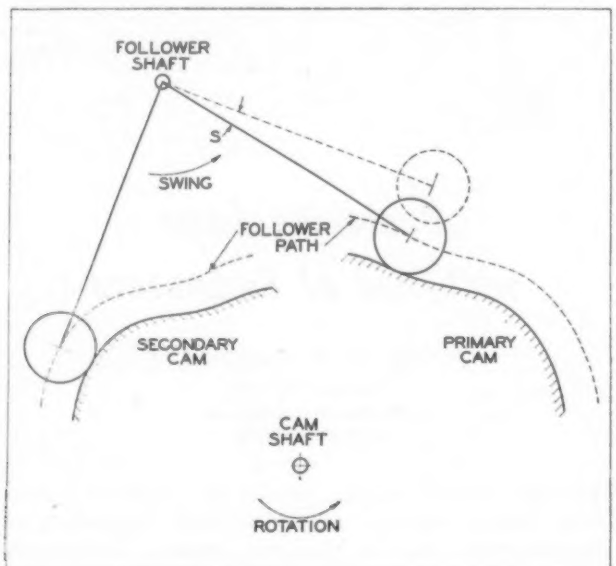
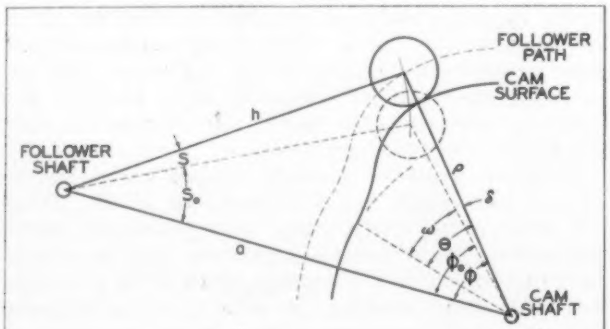


Fig. 1—Above—Complementary cam and follower arrangement utilized in weaving machine

Fig. 2—Below—Angular relationship of cams during rise



To illustrate the simplicity and speed of calculation, let us follow the necessary steps in computing a typical pair of complementary cams. These are plate cams on the same shaft whose separate followers are attached to a common fork which drives a second shaft. The cam curves are reversed so that one cam drives the fork in one direction while the other cam drives the fork in the opposite direction. The arrangement is shown in *Fig.* 1. The machine motion analysis establishes the angular location of the follower fork for particular angular positions of the camshaft, and generally, demands only a definite swing of the follower fork to occur in a specified range of cam rotation. A modified constant acceleration or trapezoidal acceleration is used to obtain the swing or rise, S. Discussions of this and other methods from dynamic view points are given by C. N. Neklutin* and D. A. Stoddart†. While other forms may be theoretically more favorable, in practice on production cams the results are comparable because of tolerances.

The data that must be calculated are the polar co-ordinates, ρ and θ, of the follower centers for complete rotation of both cams. These values are later adjusted by appropriate constants to fit the needs of the master cam in the cam cutting equipment. The first step is to obtain point by point solutions for the displacement in the sections of rise or fall of the follower. The displacement equations for the modified constant acceleration used are given in the appendix. Considering the displacement, s, at a point as being the angular swing of the follower shaft (*Fig.* 2), the radial co-ordinate, ρ, of the follower center is given by:

$$\rho = [a^2 + b^2 - 2ab \cos (s_o + s)]^{1/2} \qquad (1)$$

The angular co-ordinate of this point, however, does not agree with the angular rotation of the cam because the follower does not move radially to or from cam center. The corresponding cam angle, θ, is obtained by the following equations:

$$\cos \phi = \frac{a - b \cos (s_o + s)}{\rho} \qquad (2)$$

$$\left.\begin{aligned} \delta &= \phi - \phi_o & (a > b) \\ \delta &= (180 - \phi_o) - \phi & (b > a) \end{aligned}\right\} \qquad (3)$$

$$\omega = \frac{n}{N} \beta \qquad (4)$$

$$\theta = \omega \pm \delta \qquad (5)$$

In the above equations, β is the total cam shaft rotation that is to produce the total swing, S, of the follower shaft; N is the number of points that are to be calculated in the range β; and n is the point in question. Other values are indicated in *Fig.* 2. The sign in Equation 5 is determined by orientation and

*"Designing Cams"—MACHINE DESIGN, Vol. 24, No. 6, June 1952, Page 143.
†"Polydyne Cam Design"—MACHINE DESIGN, Vol. 25, No. 1, January 1953, Page 121.

rotational direction of the cam and follower. The sign is plus when the fork is apparently pushing the follower on the cam and minus when it is apparently pulling.

Several master decks of IBM cards have been prepared from the displacement equations, each card of a deck being punched with its point number, *n*, and the corresponding value of the displacement function, *s/k*, while each deck has a different number of points, *N*. Any deck can be subdivided to give a deck with less points by selecting every second, third, fourth, or nth card where *n* is a factor of *N*. These decks are rapidly prepared and provide a flexible advanced calculation wherein any rise can be divided into a suitable number of points. IBM cards have provision for 12 punches in a column and 80 columns to a card. Each column is used for a digit in a number and, or, for a code punch. Positions 11 and 12 (above zero in *Fig.* 3) which are also called *x* and *y*, respectively, are used for coding. The second, third, or *n*th card of one of these master decks would be coded with an *x* in a column, say column 70 for second cards, 71 for third cards, and so forth.

Assume that on one section of a cam we need a given rise to occur in 29 degrees of cam rotation with

Fig. 3—Layout of complementary cam work card for calculator

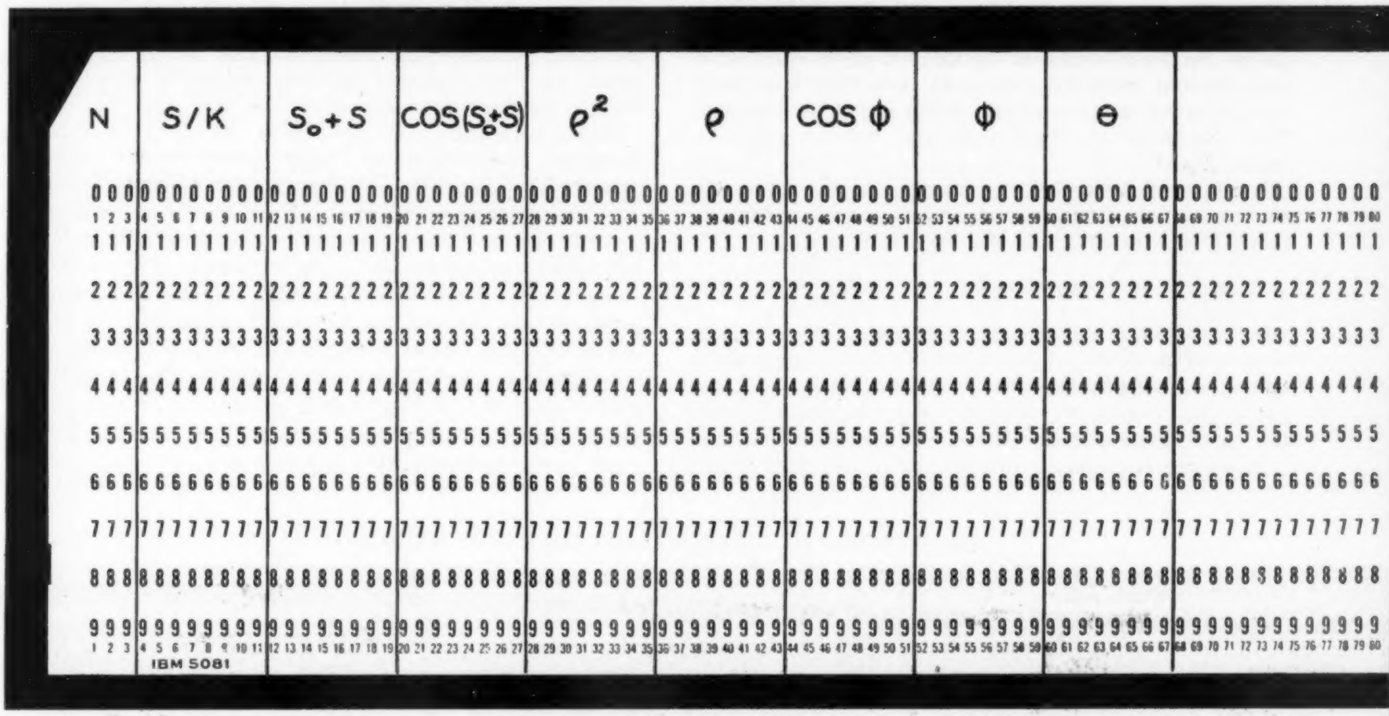
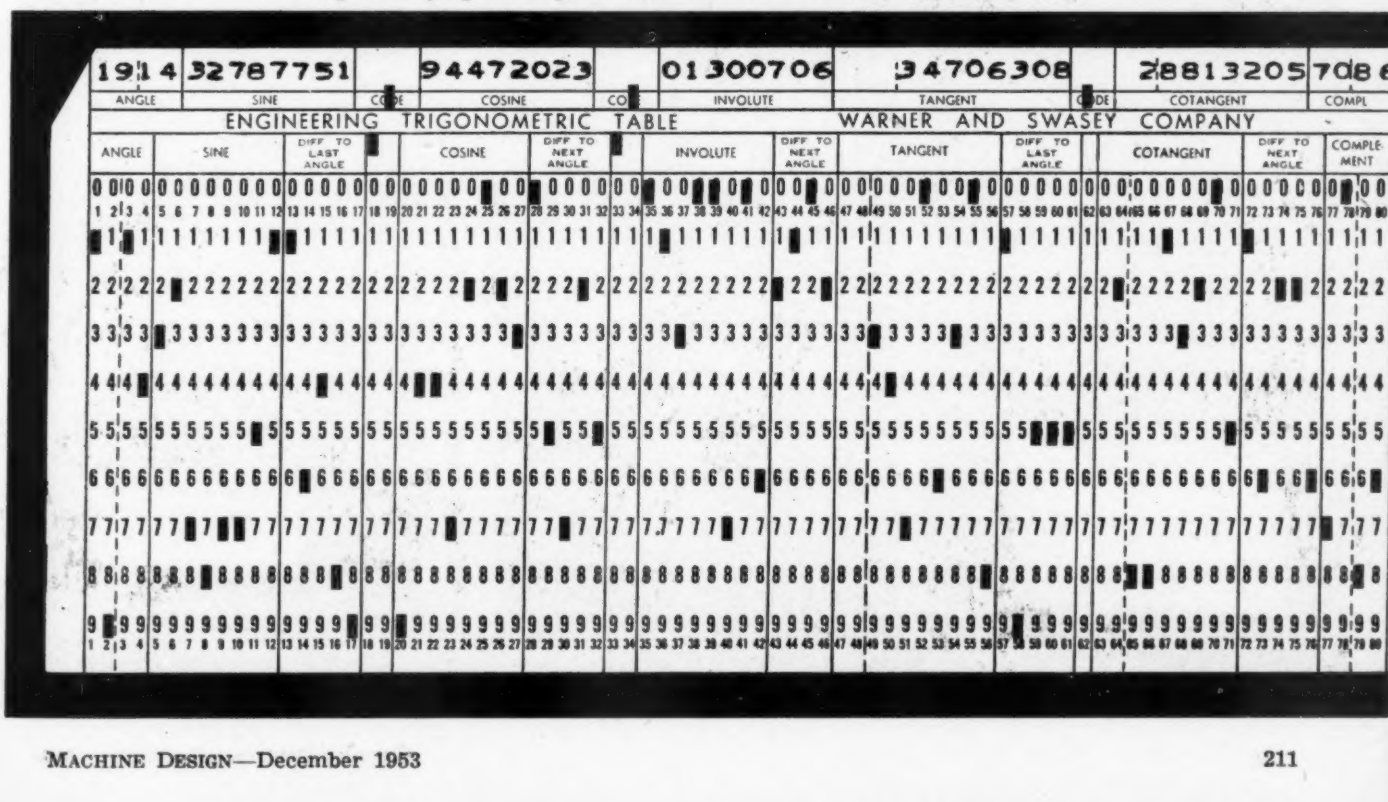


Fig. 4—Eight-place trigonometric card for cam computations



calculations at approximately half-degree intervals for reasonable master cam cutting. Selection of every fifth card from a 300-point master deck would give suitable intervals. The point number, n, and the s/k values of these cards are copied onto blank work cards, *Fig.* 3, by feeding the complete master deck and a stack of blank cards into a reproducing punch. By appropriate wiring of the control panel, the machine observes the code for fifth cards, reads only from these cards, and punches the information into the blank cards. At the same time, another code punch is put into each work card for future use. Reproducing occurs at the rate of 100 cards per minute (total cards in master deck) with a rearrangement of data columns if desired, and with a simultaneous verification of the punched results.

A pair of complementary cams may have as many as six different sections of rise or fall interspersed with dwell regions. A deck of work cards is prepared as above for each different section of each cam, each deck having different coding so that they may later be separated; at present the decks are kept separate. This completes the first step and we are ready to tackle Equation 1.

Because of limited factor storage on the Type 604 electronic calculating punch, this equation must be handled in several steps. The first is to evaluate:

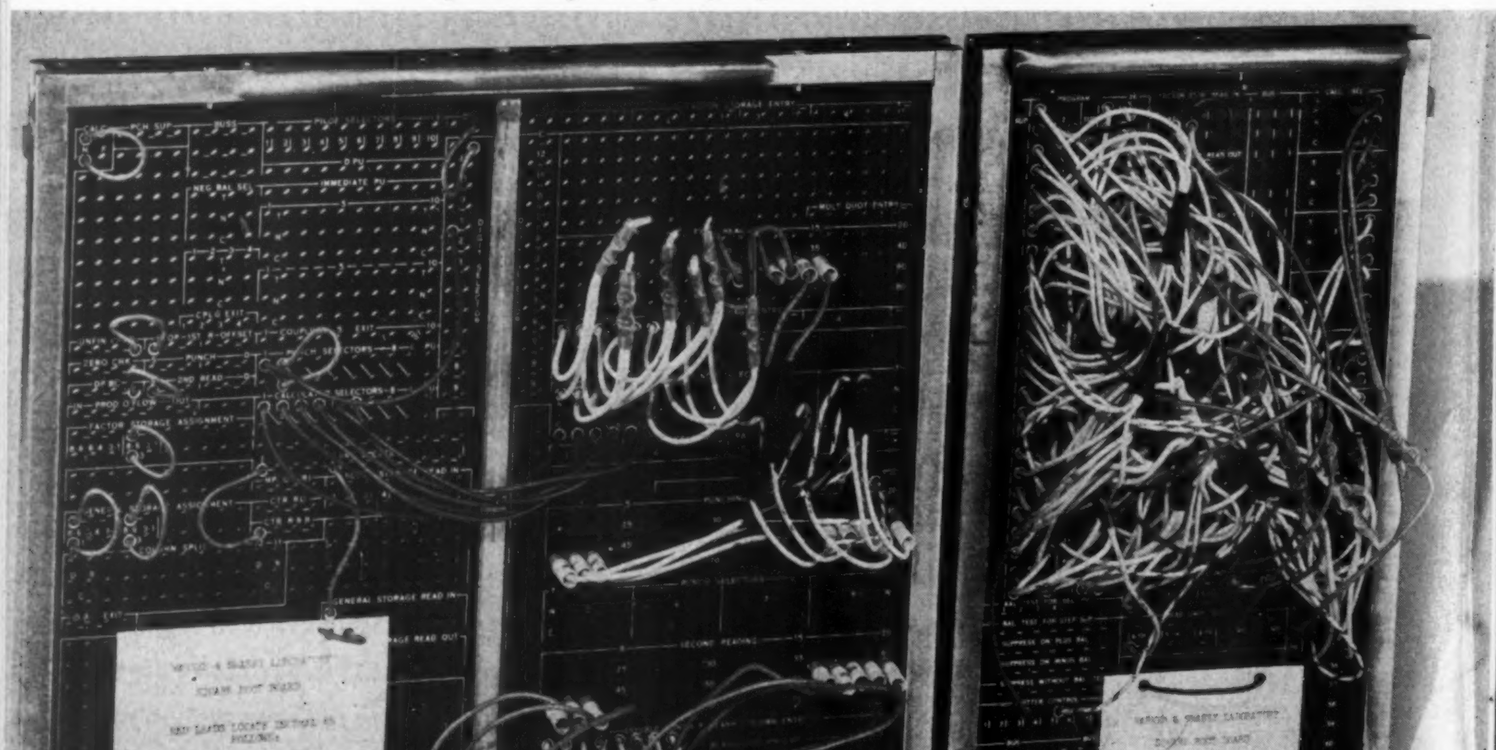
$$s_o + s = s_o + (s/k)(5\ S/N^2) \qquad (6)$$

where all but s and s/k are constants for each cam section. The decks of work cards are fed separately into the calculating punch which consists of two parts: the punch unit which reads information from the cards, generates constants, and punches computed results into the cards; and the calculator unit which receives the information from the punch, performs the desired calculation, and sends the results back to the punch. Basically, the computer can add, subtract, multiply, and divide. Numerous other features of both units permit countless varieties of code control calculation, simultaneous checking of results, generation of sequential numbers, and so forth. The computing rate is a constant 100 cards per minute.

Control of each unit is separate through individual control boards which like most IBM controls looks just about like a punch board. Desired operations are obtained by connecting correct outlets with prepared wires that have connectors similar to banana plugs. For each deck of work cards, the punch unit is wired to read s/k from column 4 to 11, emit s_o, and emit $5\ S/N^2$ and transfer these values to storage in the calculator. It is also wired to punch the computed result, $s_o + s$, in columns 12 to 19. The calculator is wired to take the values from storage and perform the multiplication and addition. The wiring of the punch unit is changed between decks to give the correct constants for that section of the cam.

Direct calculation of the cosine of $s_o + s$ with sufficient accuracy would require three separate runs through the computer with extra card columns being used for intermediate results. It is faster and easier to have the machines interpolate the cosine from an eight-place trigonometric deck of cards, *Fig.* 4. To do this, all of the separate work decks are lumped together and sorted in numerical sequence of the angle, $s_o + s$. This is done with a sorter machine which will separate 850 cards a minute into 13 categories according to the location or absence of a punch in a column. Four passes through the machine are made to sort to hundredths of a degree.

Fig. 5—Wiring for eight-digit square root on 604 calculator



The sorted work cards are then "merged" numerically with the master trigonometric deck by means of a collator. This machine will place two decks of cards into one numerical sequence, provided the decks themselves are in sequence, at the rate of over 240 cards per minute. It can also do many other functions such as selection, sorting, etc. The combined work and trigonometric cards are then fed into the computer where, by code selection, the cosine of an angle and the difference to the next tabulated cosine are read from master cards, the fractional portion of the angle lying between tabulated values is read from the work cards, and the interpolated cosine punched only on work cards. The master and work cards are then separated in the sorter by means of code in the trigonometric cards without disturbing the sequence of the master deck.

POINT NUMBER	DEGREES	DEG.	MIN.	SEC.	RADIUS
10	0 0500726	5	00	26	2 9148
11	0 0550966	5	30	34	2 9154
12	0 0601255	6	00	45	2 9161
13	0 0651593	6	30	57	2 9169
14	0 0701987	7	01	11	2 9178
15	0 0752440	7	31	27	2 9189
16	0 0802956	8	01	46	2 9202
17	0 0853540	8	32	07	2 9216
18	0 0904188	9	02	30	2 9232
19	0 0954901	9	32	56	2 9250
20	0 1005678	10	03	24	2 9269
21	0 1056518	10	33	54	2 9290
22	0 1107420	11	04	27	2 9313
23	0 1152383	11	35	01	2 9337
24	0 1209403	12	05	38	2 9363
25	0 1260481	12	36	17	2 9391
26	0 1311616	13	06	58	2 9420
27	0 1362805	13	37	41	2 9451
28	0 1414047	14	08	25	2 9484

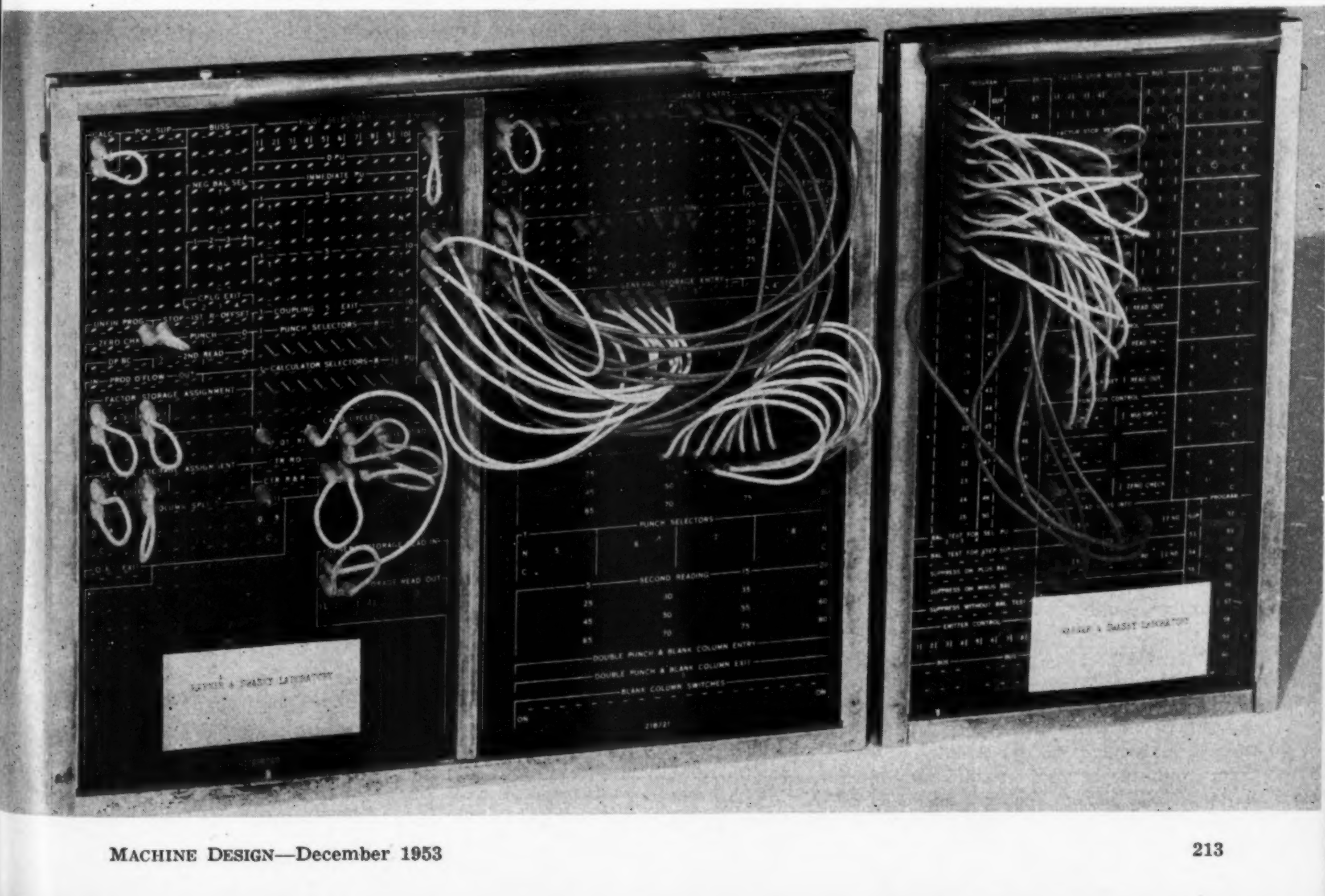
Fig. 6—Above—Sample of printed results obtained

The next step is to calculate and punch the square of the radius vector in the computer, reading the cosine of $s_o + s$ and introducing the constants a and b. Since b is generally the same for both cams, the cards do not need to be sorted into their separate sections. The square root is a lengthy process on the 604 computer and is done in a separate run. A square root is obtained by successive iterations of the equation

$$\left(\frac{B}{A_n} + A_n\right) 1/2 = A_{n+1} \qquad (7)$$

where B is the square and A is the approximation to the square root. By using 3, 30, 300, etc., according to the decimal in B, as the first approximation, A_1, and evaluating the equation three times gives an answer, A_4, which is accurate to five digits. Two more iterations gives an accuracy to eight digits. Since the control board wiring is so complex for this application, (*Fig.* 5), this board is left permanently wired. In obtaining a square root to this accuracy, the calculating portion of the machine cycle is often not quite long enough to complete the result before

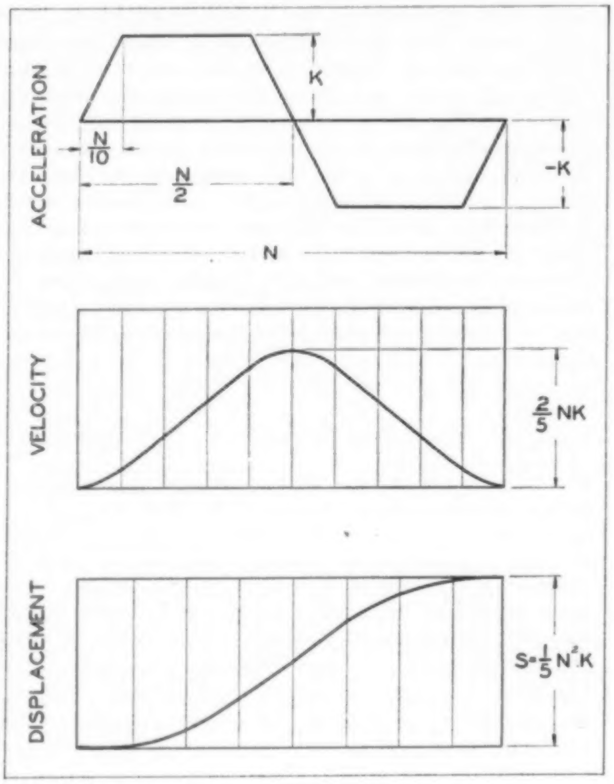
Fig. 7—Below—Wiring for calculation of Equation 6



the card is ready to be punched. As a consequence, the eight-place square roots are obtained at about 75 cards per minute.

Calculation of Equations 2 through 5 involves steps similar to those above. The procedure is to evaluate $\cos\phi$ in the calculator and then obtain a numerical sequence of $\cos\phi$ in the sorter. The work cards are then merged numerically by cosine with the trigonometric deck in the collator and the combined decks are sent through the calculator to interpolate ϕ. The trigonometric deck is separated and each work deck separated in the sorter by use of the code originally punched in them. In one run through the computer, Equations 3, 4, and 5 are calculated, punching only θ and changing the constants ϕ_o, N, β, and the algebraic sign between decks according to the cam section. The final operation is to feed the work decks separately into an accounting machine where the value of ρ and θ are read and typed in tabular form at the rate of 90 cards or lines per minute. A sample of a typed page is shown in *Fig.* 6 where the angle is printed in degrees and is also converted to degrees, minutes, and seconds. The radius has been rounded off to ten thousanths of an inch for machine settings.

Fig. 8—Cam acceleration, velocity, and displacement curves using modified constant acceleration



Although the above sounds like a formidable task just to calculate a cam, a discussion of some times and principles may aid in pointing out the speed and simplicity of operation. The first calculation attempted by engineering was the trigonometric deck which took fourteen days of planning and machine time to complete the 4500 cards. This is a relatively long time due primarily to unfamiliarity with the equipment. About six days were needed for actual machine operation while eight days were used in planning the control boards and deciding the form of the completed table. A good deal of the planning was devoted to such things as the number of cards necessary, the accuracy required, and the method of use of the completed table, for example, having the calculator simultaneously evaluate cosines of angles above and below 45 degrees with tables going only to 45 degrees.

Table 1—Basic Formulas for Modified Constant Acceleration*

Range	Acceleration	Velocity	Displacement
0 to 0.1	$\frac{a}{k} = 10\frac{n}{N}$	$\frac{v}{k} = 5N\left(\frac{n}{N}\right)^2$	$\frac{s}{k} = \frac{5}{3}N^2\left(\frac{n}{N}\right)^3$
0.1 to 0.4	$\frac{a}{k} = 1$	$\frac{v}{k} = N\left(\frac{n}{N} - \frac{1}{20}\right)$	$\frac{s}{k} = \frac{N^2}{600}\left[300\left(\frac{n}{N}\right)^2 - 30\left(\frac{n}{N}\right) + 1\right]$
0.4 to 0.6	$\frac{a}{k} = 5 - 10\frac{n}{N}$	$\frac{v}{k} = N\left[5\frac{n}{N} - 5\left(\frac{n}{N}\right)^2 - \frac{17}{20}\right]$	$\frac{s}{k} = \frac{N^2}{120}\left[300\left(\frac{n}{N}\right)^2 - 200\left(\frac{n}{N}\right)^3 - 102\frac{n}{N} + 13\right]$
0.6 to 0.9	$\frac{a}{k} = -1$	$\frac{v}{k} = N\left(\frac{19}{20} - \frac{n}{N}\right)$	$\frac{s}{k} = \frac{N^2}{600}\left[570\frac{n}{N} - 300\left(\frac{n}{N}\right)^2 - 151\right]$
0.9 to 1.0	$\frac{a}{k} = 10\frac{n}{N} - 10$	$\frac{v}{k} = N\left[5\left(\frac{n}{N}\right)^2 - 10\frac{n}{N} + 5\right]$	$\frac{s}{k} = \frac{N^2}{15}\left[25\left(\frac{n}{N}\right)^3 - 75\left(\frac{n}{N}\right)^2 + 75\left(\frac{n}{N}\right) - 22\right]$

*Where N = total number of points for calculation of rise, k = maximum acceleration in inches per point per point, $k = 5S/N^2$, S = total rise in inches, and n = point number.

The trigonometric deck could have been hand punched from existing tables, but it was faster, more accurate, and no burden on the accounting department to feed blank cards into the calculator and let it generate the sequences and series functions. The master card decks for the modified constant acceleration motion were also started from blank cards, having the calculator generate its own series of consecutive numbers, n, and simultaneously calculate the corresponding value of s/k.

The first cams that were calculated were a pair of complementary cams having five sections on each cam in which the radius was changing. The ten decks of cards for these sections represented a total of 1200 points. Mistakes and poor planning caused backtracking and unnecessary steps, but the work was completed in ten days which includes all planning and machine time. These same cams had been started on a desk calculator and it had taken 28 days to complete 90 per cent of the work on the primary cam alone. Cams of this nature are now being run in two to three days.

In general, the wiring of control boards is rather simple and, after some practice, can be done at the same time that the calculation is being mentally planned. As an example, *Fig.* 7, shows the simplicity of the punch and calculator control boards wired to solve Equation 6. One set of boards is used for all general calculations, being torn down and rewired for each calculation, while one set is kept permanently wired for square roots. Control boards for other machines are less complex.

The machines are used most advantageously for a large number of identical calculations that can be started from a sequential series. The practical minimum number of cards or points is about 25 to 100, depending upon the length of calculation and method of starting. If the initial variables have no logical sequence and must be hand punched, at least 100 cards should be used unless the calculations are quite lengthy. As an example, statistical correlation of each pair of ten variables using fifty sets of data (fifty cards) was accomplished with 26 passes through the computer taking two and a half hours with minor wiring changes between runs. This plus the half-hour spent in hand punching the variables was a considerable savings over manual methods because of the calculation length.

The additional control features of the calculator and punch unit permit numerous types of calculations in which the machine can select one of many different calculating paths by examining its own results at various stages in the calculation. A practical example of this is in the conversion of degrees to degrees, minutes, and seconds for master cam manufacture and simultaneously selecting one of three paths for values rounded off to the nearest five minutes in order to make a test cam. The latter step saves the machinist's time and reduces errors. These additional control features provide so many possibilities that most repetitious type calculations justify an investigation of accounting equipment that is available.

Appendix: The basic formulas used for modified constant acceleration have been derived from a trapezoidal acceleration diagram, *Fig.* 8, which also shows the resulting velocity and displacement diagrams. During the first tenth of the rise time period, the acceleration increases linearly from zero to the maximum. It remains constant for the next three-tenths, and then decreases linearly to zero during the next tenth. The last half of the time period has exactly the same shape but negative values of acceleration.

The resulting equations for acceleration, velocity, and displacement are given in TABLE 1.

Systematic Correlation of Motions

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A SYSTEMATIC method for correlating accurately timed relationships of machine motions has proved invaluable in solving the problems of design in the Warner and Swasey Sulzer weaving machine. In this machine—the simple eight-harness cam loom with single package filling—there are 24 different cam motions, 57 or more accurately made cams and 89 established timing factors. These motions are tied together positively by gears, linkages, splined shafts, and fixed-position couplings in many instances to fractions of a degree of rotation. The addition of such operations as filling mix, color mix and dobby motions to this machine add materially to the complexity of timed motion study. The basic machine timing diagram has been selected for purposes of this discussion and will serve amply to illustrate the method employed.

As looms have advanced from hand to power to automatic the three basic motions—shedding, filling and lay motion or "beat up"—have not changed but complementary motions for creating designs have introduced the complications of accurate timing. Timing of the conventional looms is regulated by adjustable means under control of skilled mill mechanics, called fixers, rather than inbuilt precision manufacture and assembly. Made of iron, wood, leather, and fabric with co-ordinated simple motions, the present day automatic loom is a truly ingenious machine capable of handling intricate fabric designs, but is bound by tradition and personal skill to a slow pace of development.

The weaving machine, however, is a precision proposition. Skill is transferred to the machine both from the weaver and fixer standpoint. This, then, is the basic difference between the loom and the weaving machine. A conventional automatic power loom and the new weaving machine are shown in *Fig.* 1. In the late thirties Sulzer Bros. of Winterthur, Switzerland, decided, on the basis of previous textile experience, that something should be done to make the loom comparable to other modern machines. Basic design approach would require: Transfer of skill to the machine; removal of temperamental factors of adjustment; co-ordination of motions with precision manufacture; use of unit construction instead of overall assembly; removal of the gyrating effects of heavy lay members and picking mechanism; and reduction of mass to increase speed.

Many attempts have been made by various individuals and companies to accomplish these objectives. Many of the ideas incorporated in the present Warner and Swasey Sulzer Weaving Machine have had their counterparts in other designs but never in the overall successful combination that this machine presents. The approach to this design has been that of

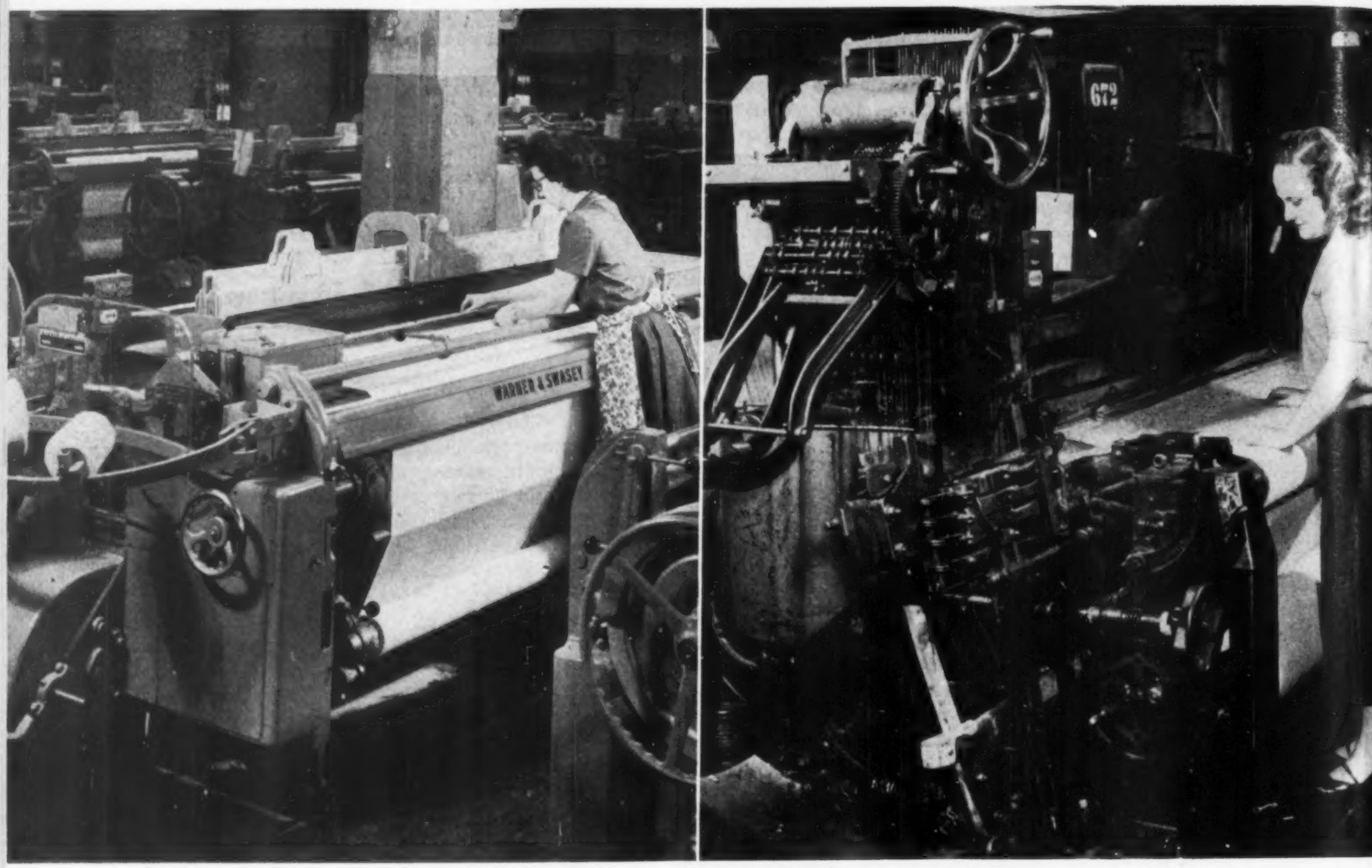
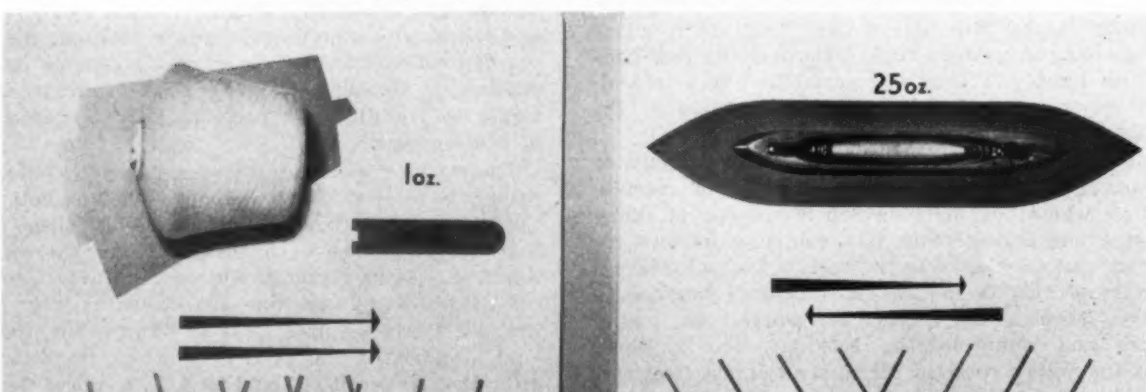


Fig. 1—Above—A conventional automatic power loom, right, and the new automatic Warner & Swasey Sulzer weaving machine

Fig. 2—Below—New method of weaving, *a*, with small shuttle and conventional method, *b*, which uses a large shuttle containing the wound bobbin



the engineer with mind free of the traditions of weaving art and individual skills.

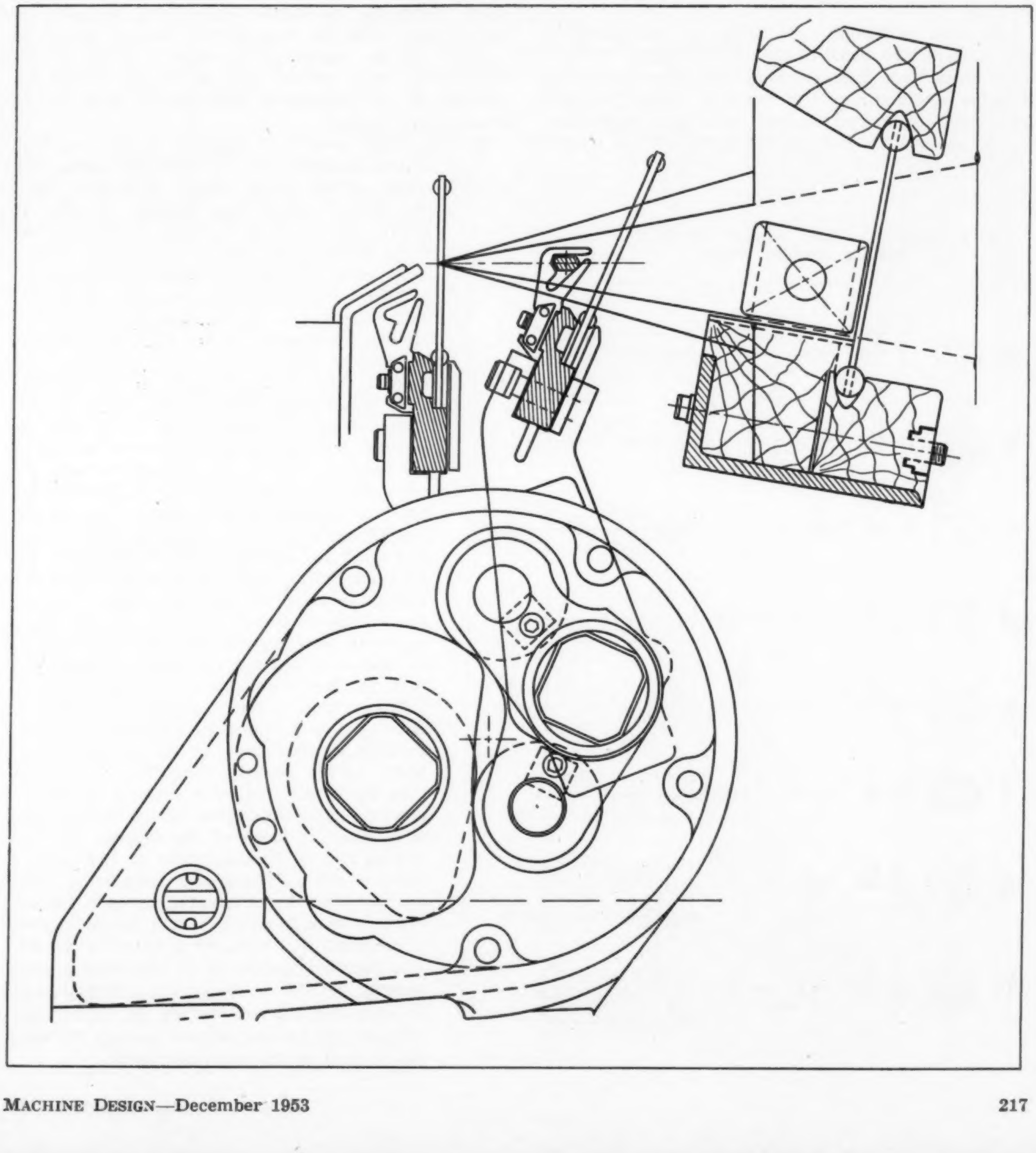
Accepting the three basic motions of weaving, viz., shedding, filling and lay motion, the radical point of departure for the weaving machine lies in the method of filling. This is illustrated by *Fig.* 2 which shows conventional method vs weaving machine method. In the conventional method, the filling yarn is wound on a small bobbin. The bobbin is mounted in the shuttle and the shuttle thrown back and forth to form filling picks in a continuous strand. In the weaving machine method, the package of filling yarn is wound on a cone or cheese, mounted on a stationary bracket outside the picking area. The yarn is transferred to a small shuttle of steel with a gripper spring by a shuttle feeder. The shuttle is thrown, with a trailing strand, across the lay bar in one direction only and returned empty by conveyor. The construction of the filling is by filling pieces rather than a continuous strand.

MOTION CORRELATION

This illustration also indicates certain factors related to the speed. The conventional method uses a shuttle measuring up to 17½ inches long by 2 inches wide and 1½ inches deep, weighing up to 25 ounces, of which, 1½ ounces is the filling package. The weaving machine method uses a shuttle 3¼ inches long by ½-inch wide by ¼-inch thick weighing 1 ounce with a filling package limited only by the cone winder in a mill. Usually 4 to 6-pound packages are wound. The shuttle carries only its trailing strand of yarn.

The small shuttle also introduces another factor of speed in that it reduces shed size and lay motion. In conventional looms, for equivalent widths to the weaving machine, the laybar with its attached picking and receiving boxes may weigh up to 280 pounds. The laybar of the weaving machine, due to separa-

Fig. 3—Laybar of the weaving machine with its mechanism and shuttle, left, compared to that of the conventional loom, right



MOTION CORRELATION

tion of picking and receiving boxes, weighs only 38 to 40 pounds, complete with reed and guide teeth. This is shown in *Fig.* 3. The net result in speed due to the change of filling method is an increase of 85 to 100 per cent.

The change of filling principle from continuous strand to individual filling strands introduces a whole new series of related motions not found in a loom. *Fig.* 4 shows diagrammatically the elements involved in laying a filling strand. Depicted are the yarn package, guide, filling brakes, tensioner, shuttle feeder, shuttle, yarn grippers and yarn cutter. The entire cycle of laying a filling strand complete with beat-up motion takes place at the rate of 240 per minute, or one quarter-second per cycle. At the point of firing the shuttle there are six to eight elements all arriving at a critical point within fractions of a degree. Under these conditions it is no longer possible to leave the timing adjustment in the hands of a fixer. Accurately controlled timing must be built into the machine.

The chart of timing correlation for the machine, *Fig.* 5, forms the basis of the entire design. It predicts cams and linkages and establishes limits of variation and accuracy requirements. These motions are correlated in the machine by a graduated dial, *Fig.* 6, secured to the driven member of the clutch shaft which makes one turn per cycle. With this dial set on 360 degrees, all locating points, keyways, dropped splines, and couplings are on top dead center.

In the basic timing diagram for the machine as indicated in *Fig.* 5, there is across the top the 360-degree cycle, divided into ten degree increments. The left-hand columns of the diagram list the motions and their graphic symbols. To the right of the diagram is a summary of motion which lists function, start and finish of motion, and motion angularity. At the extreme right of each motion column on the usual diagram are notes relating to critical points within the rise and fall of a cam.

The chart lists 32 motions. However, some of these are repeats in that a motion may be broken down into horizontal and vertical components. Similarly, for instance the motion of shuttle expeller is timed for three different relationships to the conveyor according to the position of the receiving box.

A further point in connection with the construction of the diagram is that it is not sequential with regard to a series of motions. The sequence of numbers, instead, follows the construction of the machine. Items 1 through 14 involve the picking box side of the machine, with the exception of Item 4; Items 15 through 22 the receiving box side; and Items 23 through 27 both picking and receiving. Items 28 through 32 are transverse elements of warp letoff, shedding and laybar.

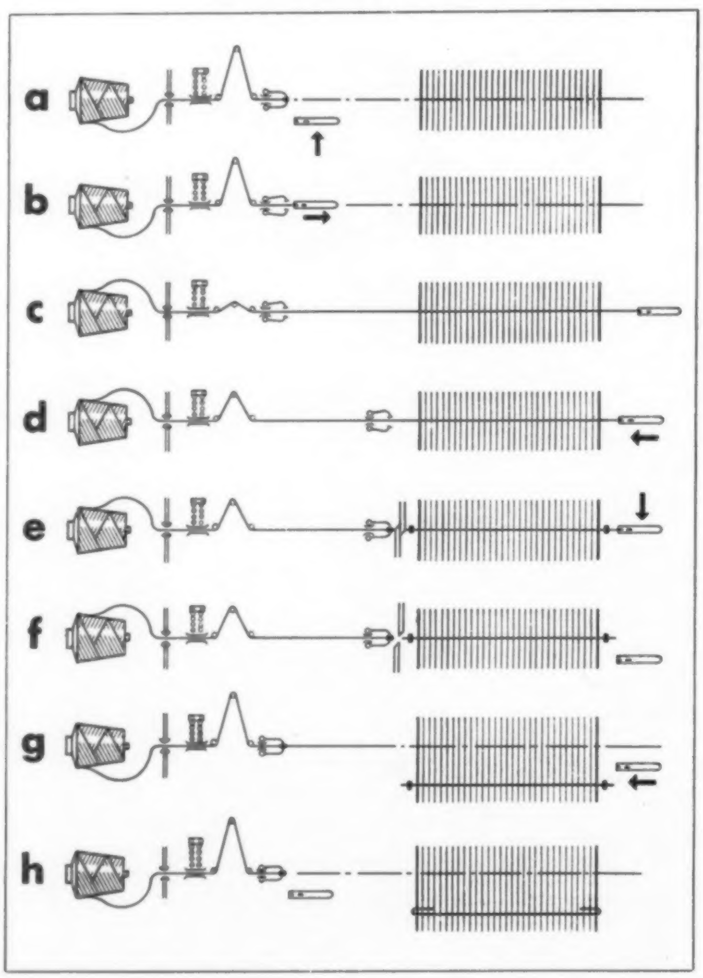
To follow the method of diagram layout, one key motion sequence will be discussed. Since the filling action is the major difference from present-day looms, the diagram in Fig. 5 shows this group of motions relating to the filling strand magnified.

The shuttle with its trailing strand of yarn is thrown from the picking box through guides on the lay bar to the receiving box, while the lay bar is stationary, by the mechanism shown in *Fig.* 7.

Here, then, on the diagram is the starting point for interpretation of the laying of the filling strand. Consider Items 11, 12, 13, 15 and 29. Starting at 120 degrees of the cycle for reasons of prior loading of the yarn in the shuttle gripper spring, No. 11—the picking lever—is unlocked at 120 degrees; No. 12, the torsion rod, acting as a spring, is fully loaded and ready for release at 120 degrees; No. 13, the picking lever trips at 120 degrees and the shuttle is fired through the guide teeth of the lay bar to the receiving box in less than one-tenth of a second. The action is similar to that of a catapult throwing a missile except that in this instance the missile is guided.

Flight time is not shown directly on the chart, but is indicated by a sequence of happenings, first indicated by No. 15, shuttle detector in the receiving box. This line shows that the shuttle has been detected in the receiving box at 280 degrees for an elapsed time from point of firing of 160 degrees. The lay motion No. 29 became static at 120 degrees for the shuttle firing and remains so until 360 degrees. The differential between 280 degrees of shuttle detection and the 360-degree relationship of the laybar is taken up by several functions having to do with shuttle positioning, second detection and essential time to stop the machine before the laybar can operate and thereby prevent damage to warp yarns, reed, shuttle and guide teeth.

Fig. 4—Series of related motions involved in the weaving machine sequence



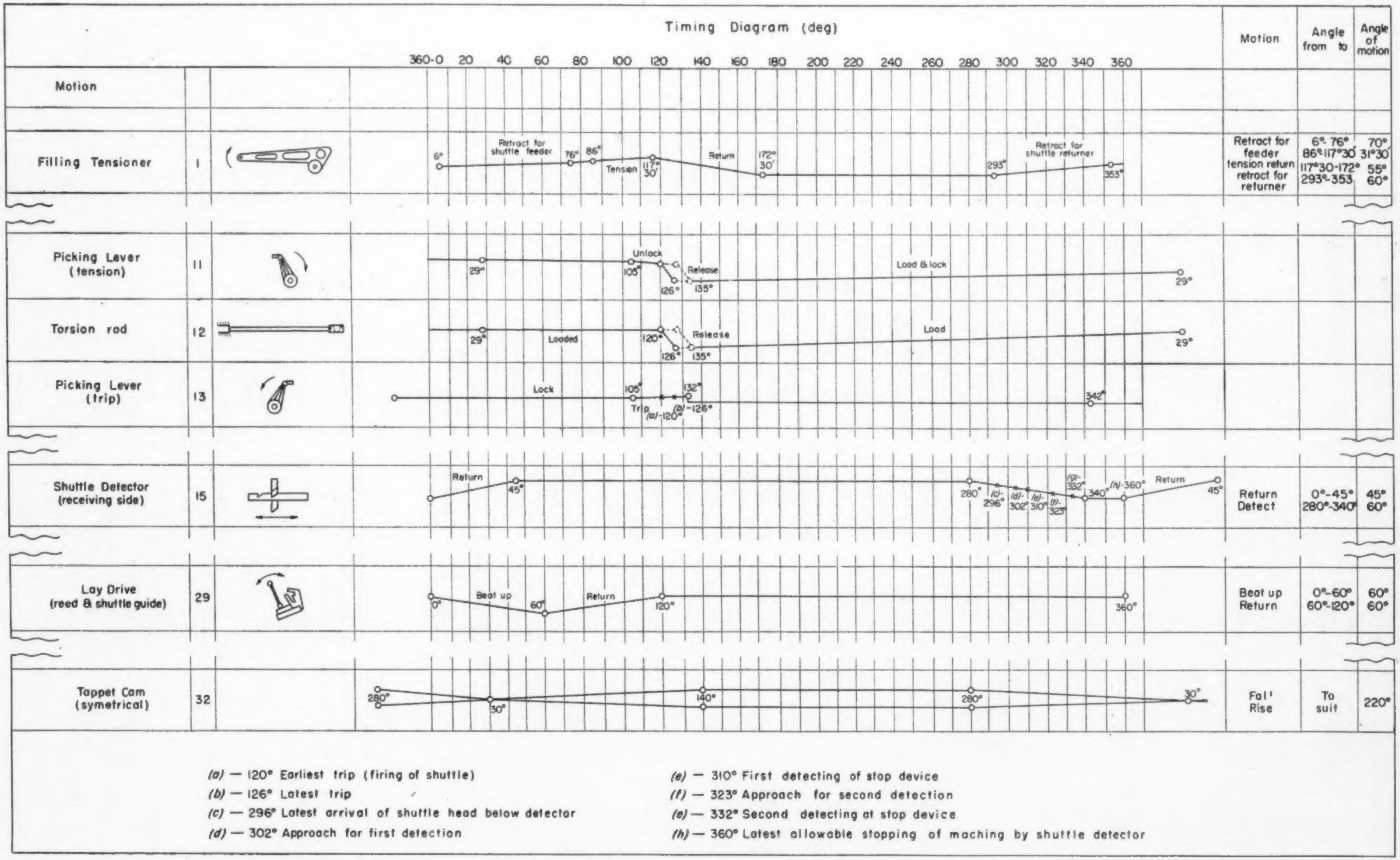
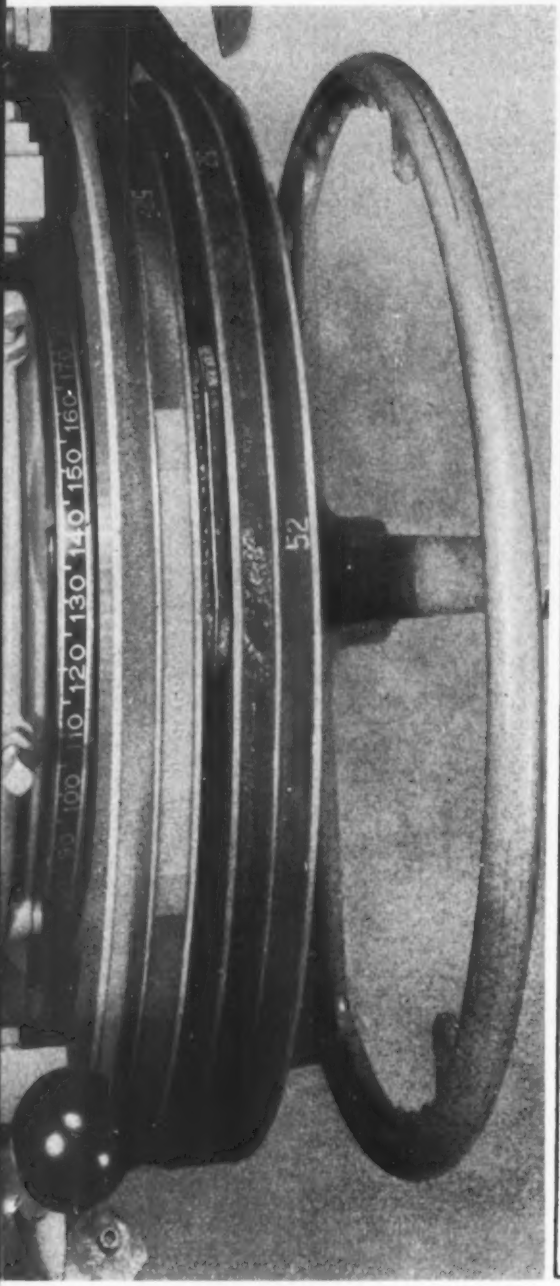


Fig. 5—Excerpts of the basic machine chart which diagrams the 32 motions and correlates the precise instant at which each step occurs



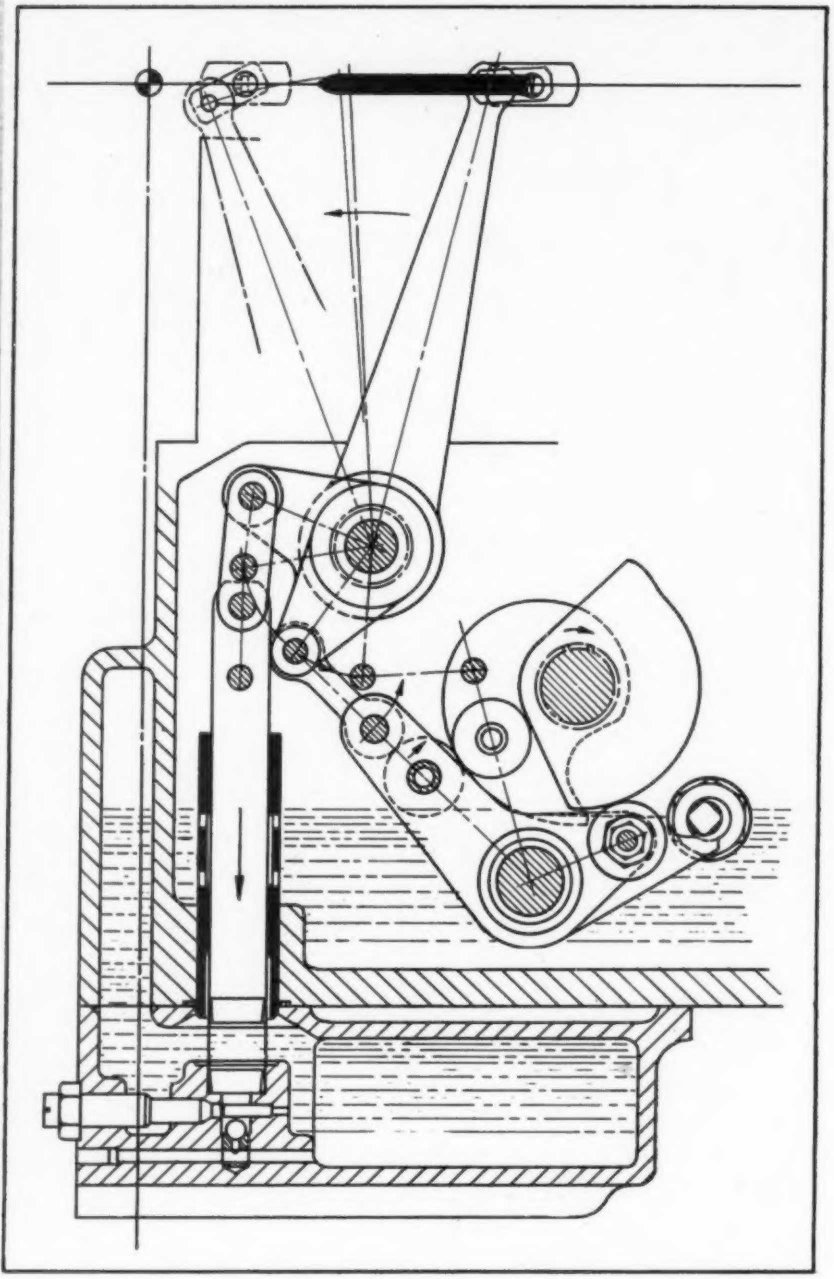


Fig. 6—Graduated dial control of the machine permits hand setting and servicing

Fig. 7—Shuttle throwing mechanism of the machine correlated in the magnified portion of the motion chart

Each element of the 32 used on the diagram shows the various points connected by horizontal as well as sloping lines. Horizontal lines represent dwell periods and the sloping portions indicate rise or fall of cams. As this is the basic diagram the rate of rise and fall is not represented. Other diagrams of the individual motion are made for this purpose.

On some elements, in addition to the main locating points there are others marked on the rise and fall sections. This is brought out especially on Item 15, the shuttle detector. These additional points are critical and set up limitations of contour accuracy, processing methods, tooling designs and inspection points.

There are many other interesting relationships of motions in this machine but enough have been cited to show how the chart is interpreted and used. The diagram or chart correlating all of these motions forms the basis for the design of the entire machine. It indicates cam setting, cam throws, limits of accuracy that must be held in manufacture, dictates processing methods, tooling requirements. Where, through reasons of weaving machine usage, changes of design have to be considered, as for instance filling mix and color mix, the chart acts as a guide to determine their feasibility and ultimately dictates limitations of speed.

From this chart also, data can be tabulated for use by assembly and inspection departments. The use of the chart in connection with the graduated dial can be used for troubleshooting, for servicing, for demonstration, and for training of personnel, not only in the factory and engineering departments but also in the weaving mill for operators and machine maintenance.

□ □ □

Synchronized Load Control

. . . meets varying demand on hydraulic power pump

By E. W. Theilig
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Reciprocating Pump Div.
Worthington Corp.
Harrison, N. J.

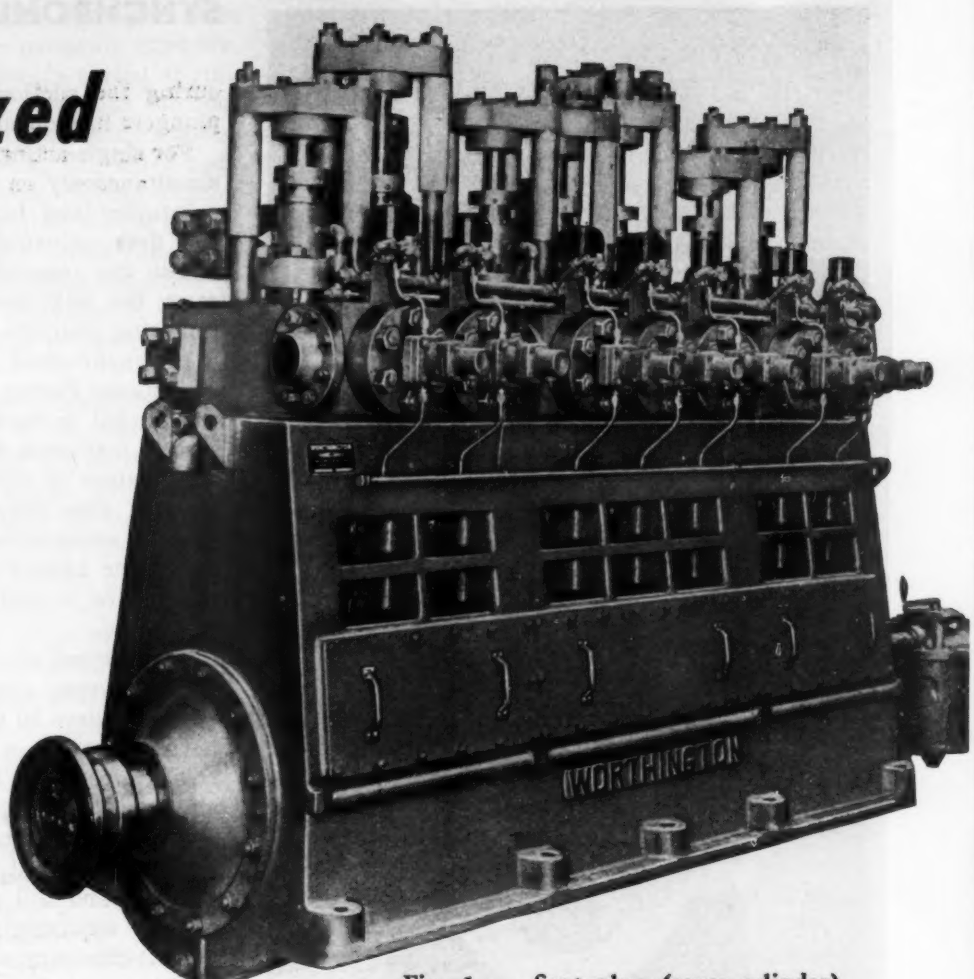


Fig. 1 — Septuplex (seven-cylinder) pump supplying hydraulic power to large hydraulic presses built for Air Force program. This unit can deliver 233 gallons per minute against 4500 psi

HIGH intermittent pressure and volume demands on the hydraulic power system are typical of large-press operation. The enormous capacity of units in the Air Force heavy press program posed unusually severe problems of meeting demands with a power system of reasonable physical dimensions. Closed-die forging presses of 25,000 to 50,000 tons capacity and extrusion presses up to 20,000 tons are included in the program.

The power system chosen employs accumulators with constant-speed, constant-displacement reciprocating pumps, *Fig.* 1, having a unique synchronized unloading and loading control which is the subject of this article.

As in any accumulator system the hydropneumatic accumulator stores sufficient energy to meet any expected fluid demand of limited duration. Delivery from this accumulator depends on the expansion of high-pressure air, from which it follows that there is a predetermined pressure differential. The effective capacity of this type of accumulator is the amount of water which is displaced by the expansion of the air through the specified pressure range.

In order that the system may meet the variable and intermittent characteristics of the fluid demand, the motor and pump run continuously, the pump being unloaded and loaded by maintaining the suction valves either inoperative or operative. In a reciprocating pump each plunger flow starts from zero at the beginning of the stroke and increases smoothly to a maximum at midstroke and then decreases smoothly to zero at the end of the stroke, the flow from three or more plungers combining to give a total flow with only minor variations. In synchronized unloading, the suction valves are made inoperative by lifters which hold the valves open during the discharge stroke and prevent pressure build-up in the cylinder. Each lifter completes its stroke

SYNCHRONIZED LOAD CONTROL

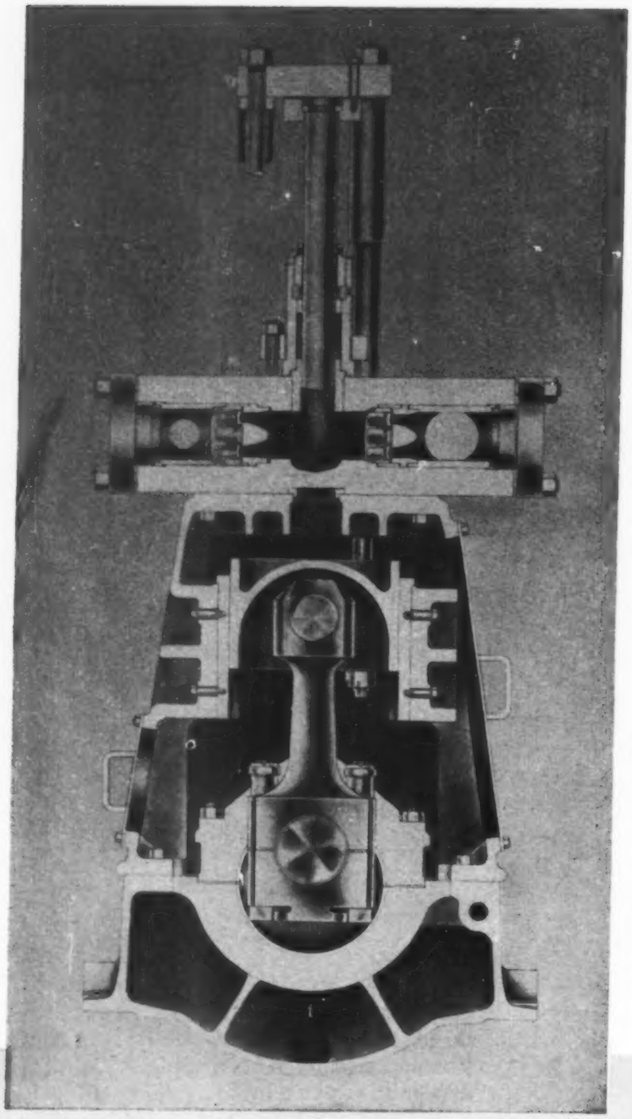


Fig. 2 — Cutaway of pump showing arrangement of single-acting vertical plunger with suction and discharge valves

during the suction stroke, thus unloading all the plungers in sequence.

For single-acting triplex pumps, two plungers are simultaneously on the discharge stroke—three for quintuplex and four for septuplex pumps. When the first unloaded plunger starts its discharge stroke the remaining one, two or three plungers carry the total flow smoothly down to zero, as all following plungers are also unloaded.

In synchronized loading, the lifters are retracted in sequence during the suction stroke and the plungers, loaded in turn, build the flow smoothly from zero to maximum without shock.

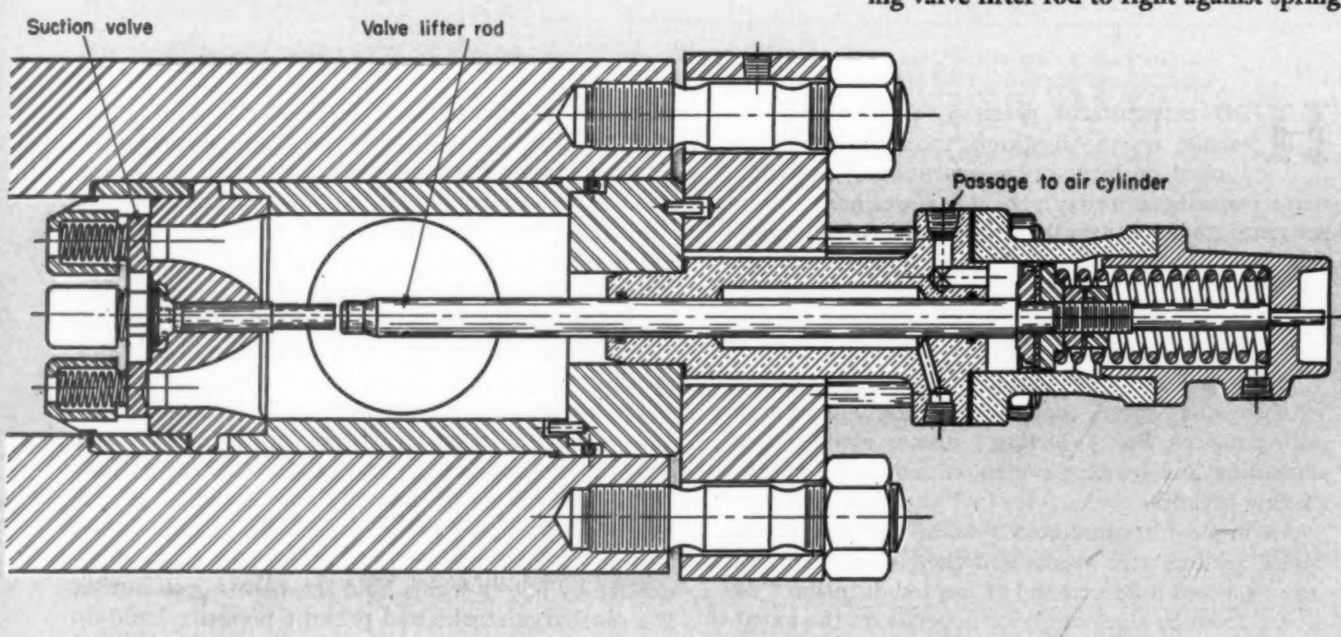
A feature of this pump control is the short element of time required for unloading and loading. This is accomplished within one pump revolution for either unloading or loading. Maintenance of this device is negligible, and hydraulic shocks are eliminated.

The vertical short-stroke pump, *Fig.* 1, is driven by a 300-rpm synchronous motor. This type of pump requires 50 per cent less floor space than the horizontal type—a major item since in some of the larger press installations from six to twelve units are required for operation.

The new high-pressure pumps for the heavy press program are built and designed to operate against 4500 psi and will deliver 233 gallons of water per minute, requiring 665 brake horsepower, with an overall efficiency of 92 per cent.

This type of pump has the liquid end designed

Fig. 3 — Cross-section through suction valve and lifter. Compression spring holds suction valve off its seat except when air pressure acts on air piston, pushing valve lifter rod to right against spring



with valve arrangement such that the liquid flows in a straight path through the pulsation chamber, *Fig.* 2. The valves are of the double ported or ring type, which allows better outlet area than the conventional wing type, consequently allowing a lower lift and better assurance of having the valve back on its seat at the end of the stroke. Spring loading on this type of valve is most essential for the proper functioning of the valve, but the valves cannot be loaded too heavily or the head of liquid required to fill the pump to insure proper operation at the required capacity without cavitation or other adverse effects becomes too great.

The valve lifters are located at each suction valve, *Fig.* 3. The spring at the right of the assembly actuates the valve lifter rod to hold the suction valve off its seat and thus unload the pump. Air pressure applied to the pistons compresses the spring, retracts the valve lifter rod and allows the suction valve to seat normally, thus loading the pump.

Admission of air, from the plant air system, is controlled by solenoid-operated air valves located at each valve lifter. These solenoid valves operate in sequence timed by an electric synchronized distributor which in turn is timed to the firing order of the pump crankshaft. When the solenoid valves are energized they supply air to lifter cylinders, and when de-energized they exhaust air from lifters.

Electrical Distributor Times Valves

An electrical distributor is driven from the pump crankshaft at pump speed and controlled by a pressure switch or float switch on the accumulator. This distributor energizes the solenoid air valves with a timing such that each lifter rod retracts from the suction valve during the suction stroke to load the pump. For unloading, each valve lifter rod advances against the suction valve during the discharge stroke and follows the opening valve to hold it open on the suction stroke.

The rotating cam assembly of the distributor comprises a body with inner and outer cams, driven at crankshaft speed. The assembly is pulled outward by the energized solenoid at the end of the assembly and is returned inward by a spring.

Spaced at equal intervals circumferentially are actuators and precision switches. These may total 3, 5, 7, or 9 depending on the type of pump. Movement of an actuator to the right closes its switch and to the left opens the switch.

The timing sequence is obtained by the circumferential arrangement of the switches. The inner loading cam is advanced 45 degrees to allow for the time interval between the closing of a switch and the resulting motion of the valve lifter rod. The outer unloading cam is advanced 100 degrees further to obtain the action described. The switches are wired to three-way solenoid-operated air valves to admit air to valve lifters for suction valve operation.

The synchronized suction valve unloader lends itself to any application where motor-driven reciprocating pumps with accumulator system are selected to meet intermittent power demands. Advantages include these features:

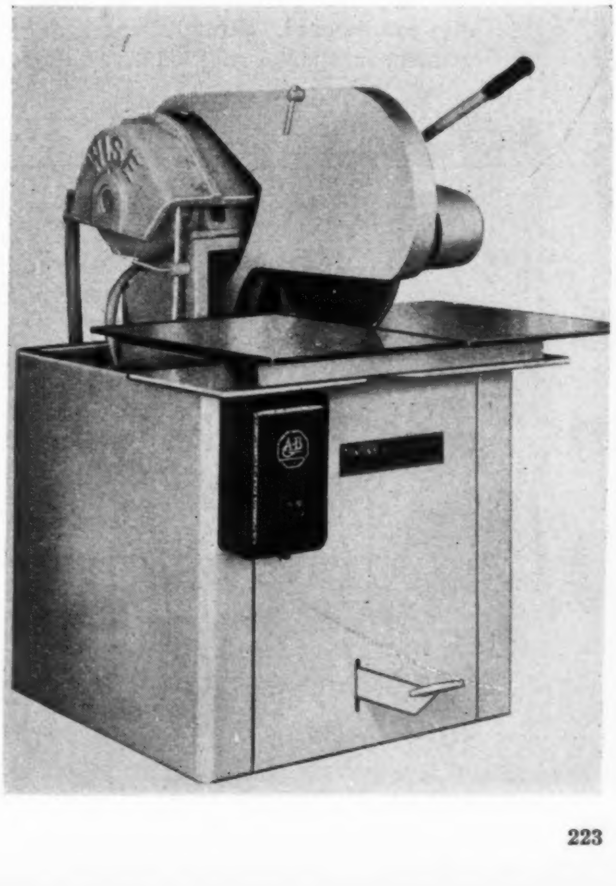
1. Minimum shock in loading and unloading, as both operations are done on the suction stroke of the pump
2. Positive synchronization of all cylinders in loading and unloading regardless of point in pump revolution when control functions
3. Pump automatically unloads on interruption of either electric or air-pressure supply
4. No loss of pressure water in loading or unloading.

Because of its versatility the system is well adapted to applications such as oil pipe lines, product lines, water flooding for oil recovery, and many other uses as well as hydraulic press work.

Vertical Saw Has Multiple Uses

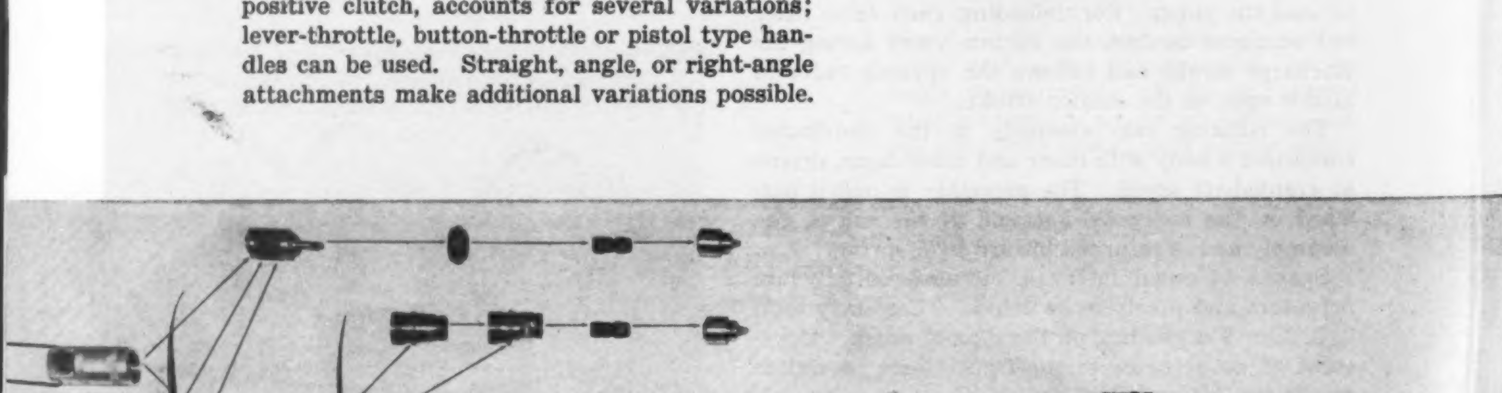
DESIGN of a new heavy-duty, all-purpose saw makes it an extremely versatile machine. Hinging the cutting head on the motor end bells provides additional throat room for handling extra-large material. Additionally, a foundry bed may be attached, cutting head locked and handle removed for cutting gates and risers from castings. Minus these modifications, the saw may be used for chop-stroke cutting of bars, pipe or shapes.

Other features of this product of the Henry H. Wise Co. are a safety guard which covers the entire cutting wheel before chop-stroke cutting starts and a heavy-duty, foot-operated vise hold-down for bars, pipe and shapes. Use of 7½ and 15-horsepower motors permits use of 18 and 20-inch cutoff wheels which make faster cuts, thereby reducing saw wear.

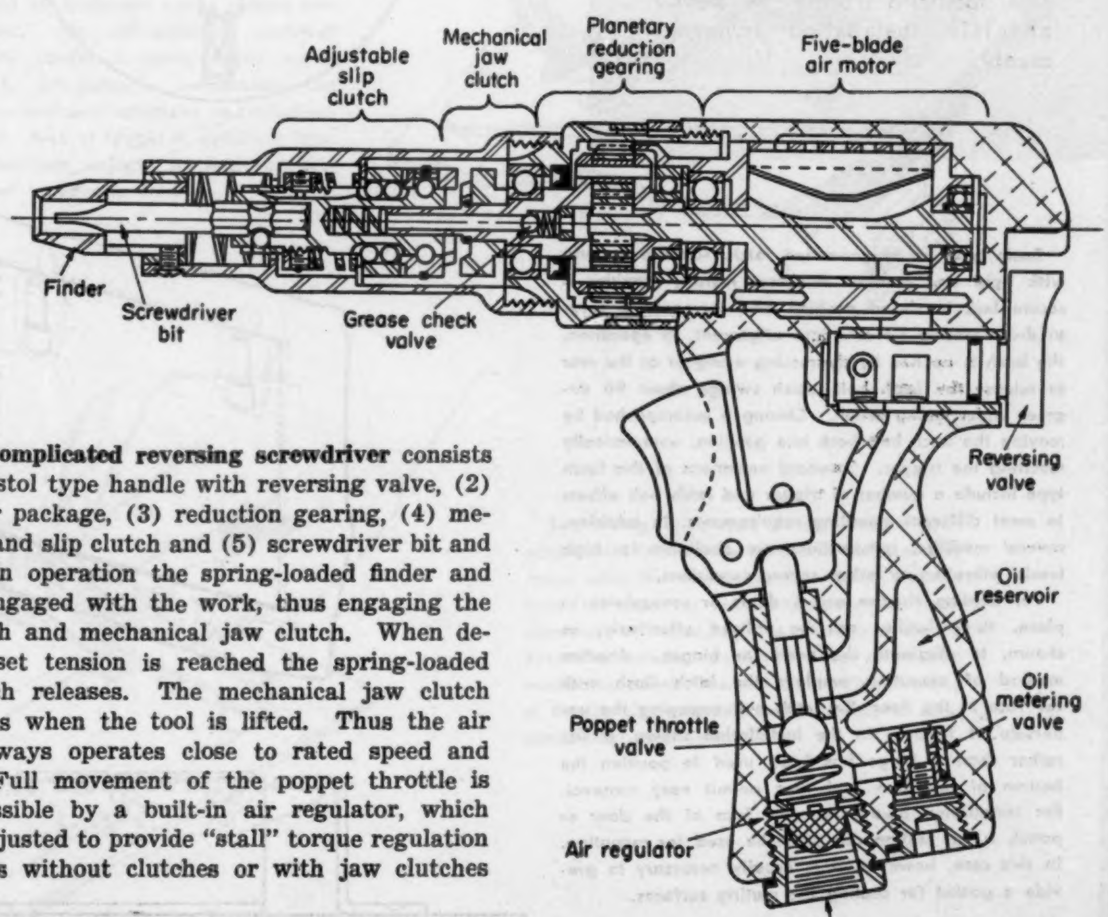


Air Tools Assembled from Interchangeable Units

OVER 100 air tool models comprising the No. 2 series of Thor Power Tool Co. are assembled from a limited number of standard subassemblies. Four basic speed ranges from 475 to 2000 rpm are available through gearing. Option of direct drive, positive clutch, or slip clutch plus positive clutch, accounts for several variations; lever-throttle, button-throttle or pistol type handles can be used. Straight, angle, or right-angle attachments make additional variations possible.



A typical simple model, for instance, consists of the air motor unit, removeable as a package, planetary reduction gearing and a collet chuck. A one-piece rotor with five blades provides a two-blade air seal for minimum air loss and maximum power. Exhaust noise is reduced by a three-stage muffler. Gear lubrication is provided by a bored spindle and spring-loaded check valve, permitting the gear case to be greased with a pressure gun and preventing loss of lubricant. The air motor is lubricated by introducing oil into the air supply from a built-in reservoir.

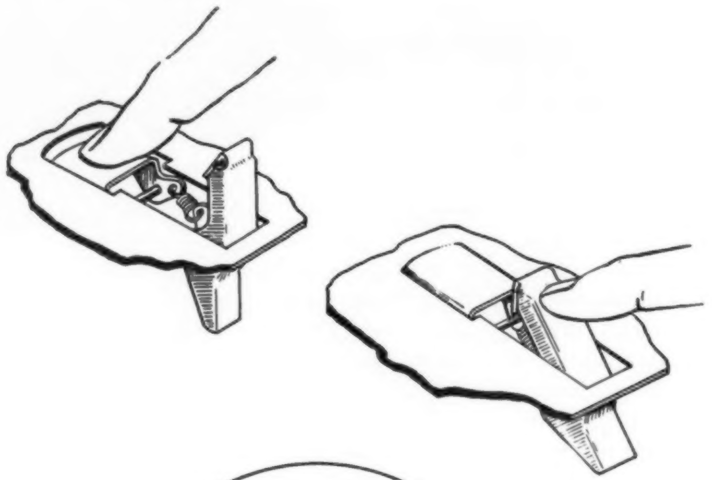


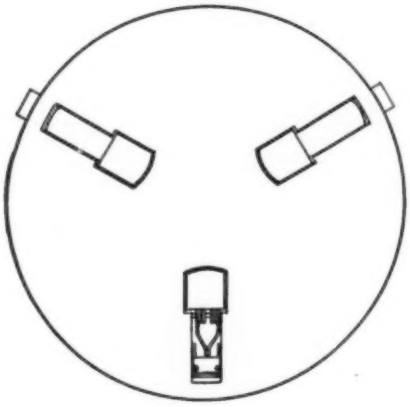
More complicated reversing screwdriver consists of (1) pistol type handle with reversing valve, (2) air motor package, (3) reduction gearing, (4) mechanical and slip clutch and (5) screwdriver bit and finder. In operation the spring-loaded finder and bit are engaged with the work, thus engaging the slip clutch and mechanical jaw clutch. When desired preset tension is reached the spring-loaded slip clutch releases. The mechanical jaw clutch disengages when the tool is lifted. Thus the air motor always operates close to rated speed and power. Full movement of the poppet throttle is made possible by a built-in air regulator, which can be adjusted to provide "stall" torque regulation on models without clutches or with jaw clutches only.

Flush Latches

DESIGN DETAILS

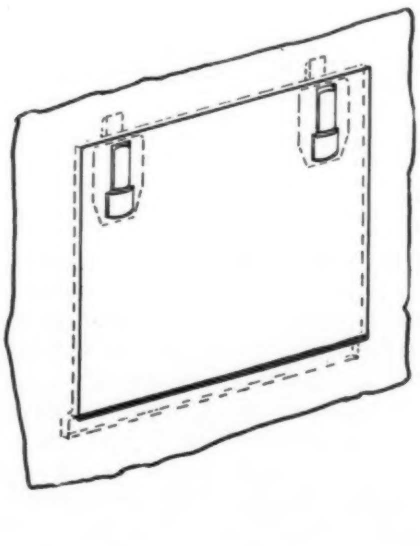
A development of the aircraft industry, flush latches today offer a wide range of practical usefulness for door or panel fastening. The combined features of flush mounting and positive operation make these fasteners particularly attractive as functional design components contributing to product appearance, performance, and safety. Available types can be readily adapted to varying sizes, shapes, and thicknesses of metal, plastic or wood. In most cases, hinges can be eliminated through a multiple installation in which the latches themselves serve as pivot connections. Typical fastening possibilities in design are shown in the accompanying illustrations on this and the following pages which illustrate some standard latch types and their application. Products of the Hartwell Co., these flush latches are also available in several modified forms to meet specific installation requirements.

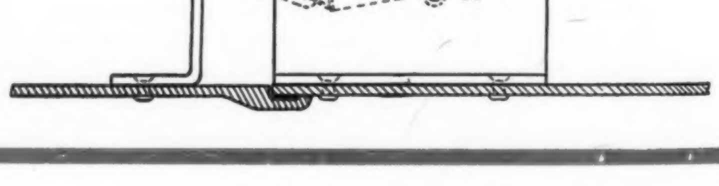




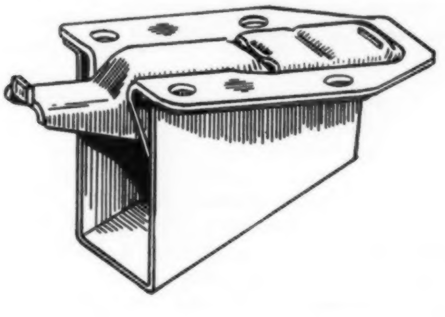
Basic type of trigger-action latch, suitable for use with light and medium thickness metals, provides a secure lock in closed position and serves as a lever to draw mating surfaces into alignment. In operation, the latch is opened by depressing a trigger at the rear to release the latch bolt which swings about 90 degrees under spring action. Closing is accomplished by moving the latch bolt back into position, automatically resetting the trigger. Standard variations of this latch type include a number of trigger and latch bolt offsets to meet different mounting requirements. In addition, several modified constructions are available for high loads, vibration or other severe conditions.

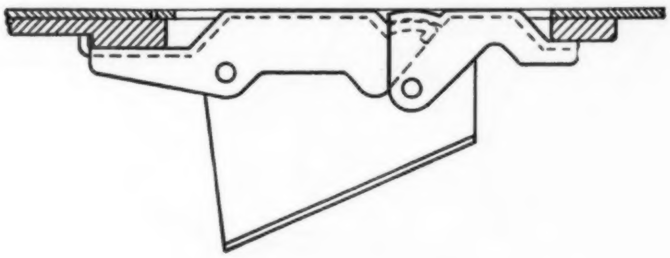
For holding circular access doors or coverplates in place, three latches can be utilized effectively, as shown, to eliminate the need for hinges. Another method of mounting employs the latch flush with the face of the door, the latch bolt engaging the underside of frame. In the installation shown, a slot rather than a hinge has been used to position the bottom of the door and thus permit easy removal. For installation away from the face of the door or panel, a hat section can often be used for mounting. In this case, however, it is usually necessary to provide a gasket for sealing the mating surfaces.





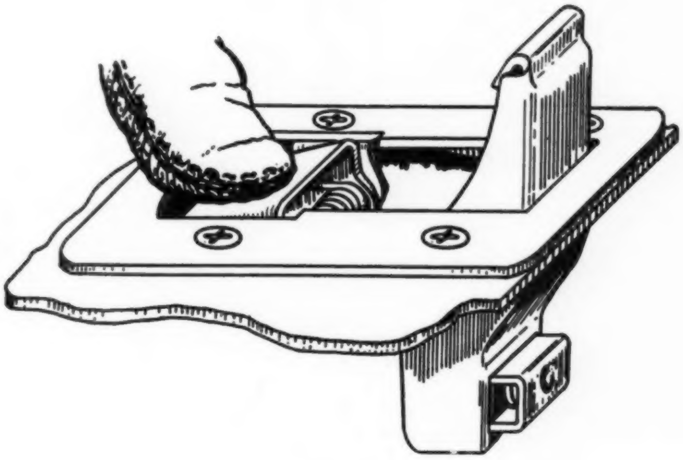
Practical solutions for design problems with standard components

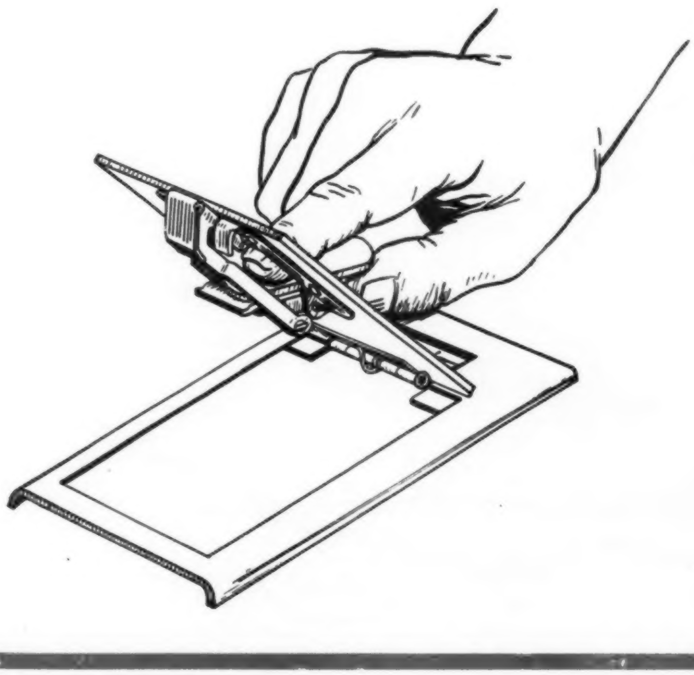




A modification of the basic type, this trigger-action latch provides a rugged shielded construction to protect the latch mechanism from damage by radiated heat or flying particles. In addition, the hook shape at the end of the latch bolt is advantageous for preventing motion of the door or panel along an axis parallel to the latch. Suitable for use where heavy load and vibration conditions exist, this design will stand up under severe service without accidental opening.

Oversize trigger-action latch for installation on large equipment is designed to withstand severe service conditions. The self-adjusting latch-bolt saddle carries a spring-loaded button which automatically compensates for frame thickness variations and misalignment. Face of trigger is especially large to facilitate opening of latch by personnel in adverse weather.



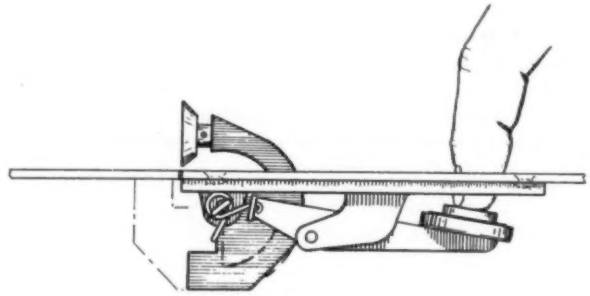


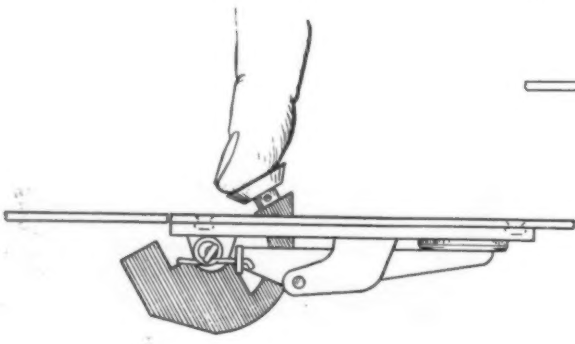
Self-closing latch type differs from trigger-action latch in that the latch bolt automatically resets when the door or panel is closed. Variations in mounting frame thickness are compensated by changing the thickness of the operating handle. Design of the self-closing mechanism permits the use of an infinite variety of handle sizes, shapes, and materials to meet operational requirements.

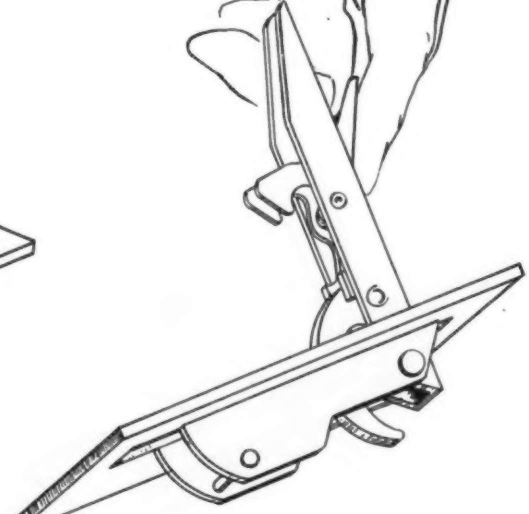
DESIGN DETAILS

A modification of the self-closing latch, this type employs two pushbuttons to replace the conventional operating handle. Opening is accomplished by a simple pushing and lifting action which acts to withdraw the latch bolt. A sturdy grip for opening the latched surface is provided by the rear button which raises as the latch bolt is withdrawn.

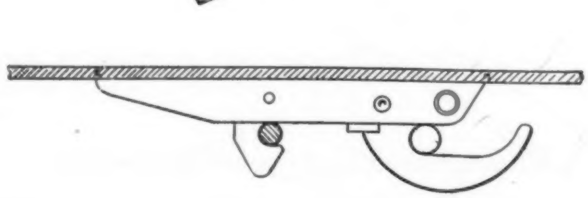
Combining the features of trigger action and pushbutton operation, this flush latch has only two buttons visible when closed. Installation is facilitated by the design which can be readily sealed for protection against splashed liquids, dust and light.



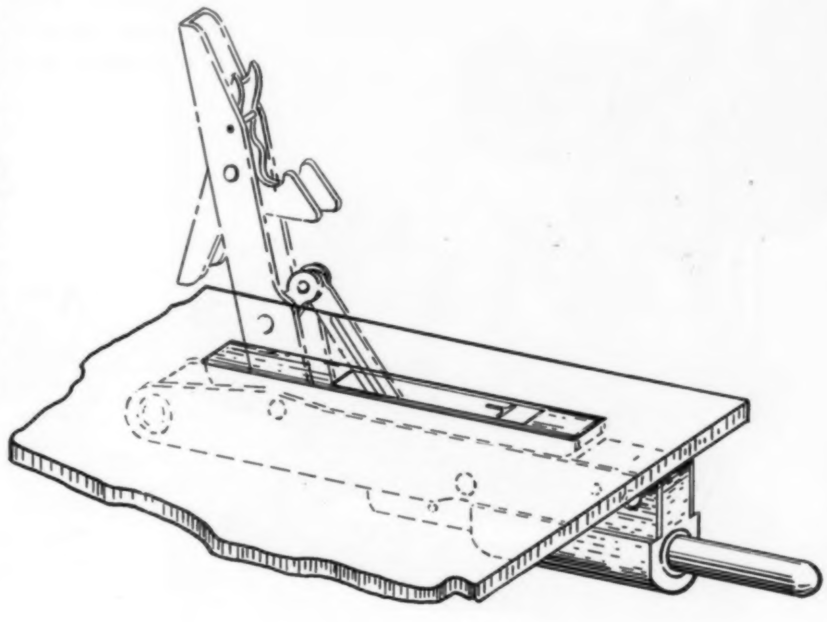


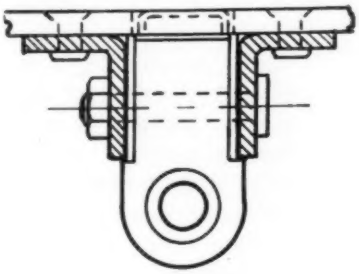


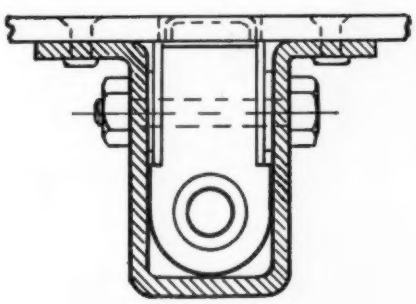
Hook type latch provides positive closing under heavy load conditions. Opening a release trigger permits the body of the latch to revolve about a point, allowing the hook to swing clear of an engaging pin. The two-pin design safeguards the latch against accidental opening and the cam take up construction of the hook provides additional force to assure positive closing. In addition, the rectangular slot mounting simplifies installation.

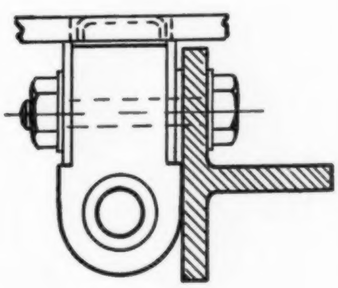


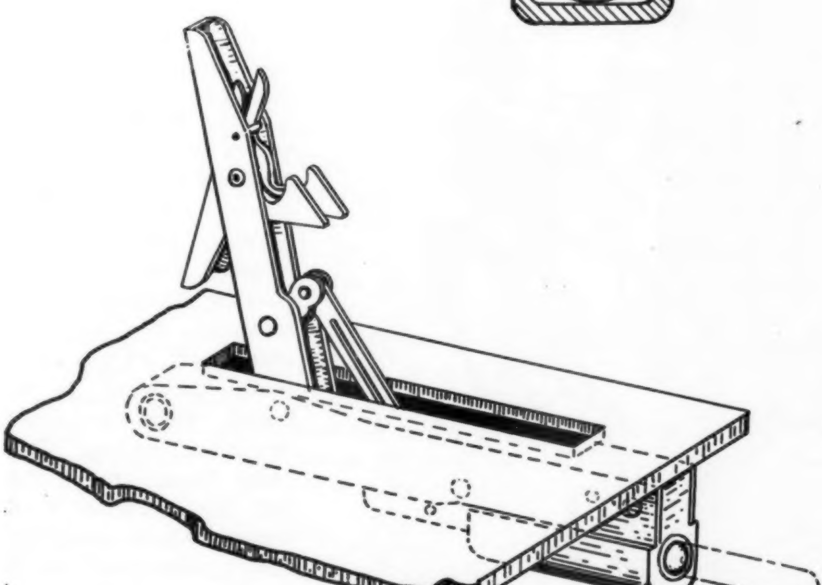
Pin latch employs a sliding bolt to assure secure fastening under severe load conditions. A double guard against accidental opening is provided by the operating lever which is itself a hook type latch. The protruding pin engagement permits the latches to be used as pivots in a multiple installation, eliminating the need for hinges and at the same time facilitating complete removal of the panel or door. In addition, the tapered pin end provides a "draw-in" action to assure complete closing. Latch installation is facilitated by the design which requires only a rectangular cutout and can be mounted by several different methods as shown.

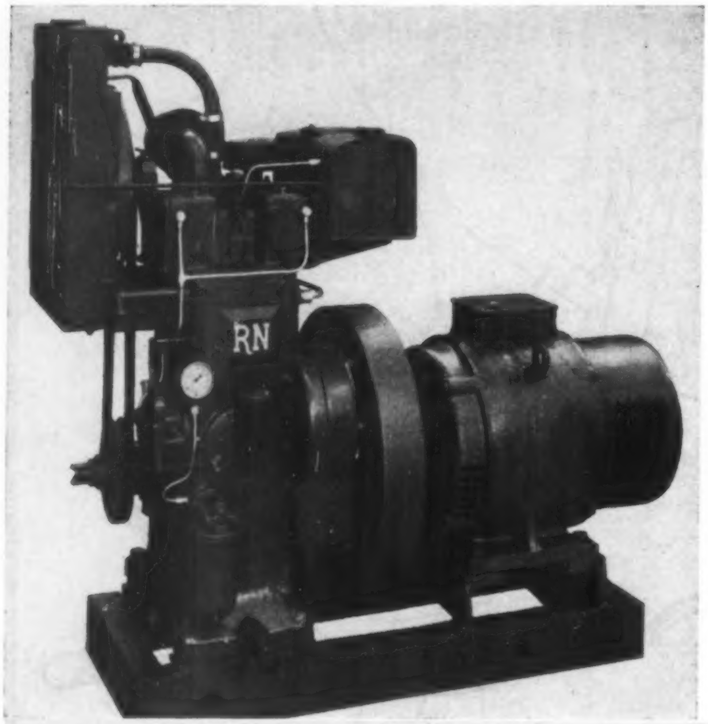






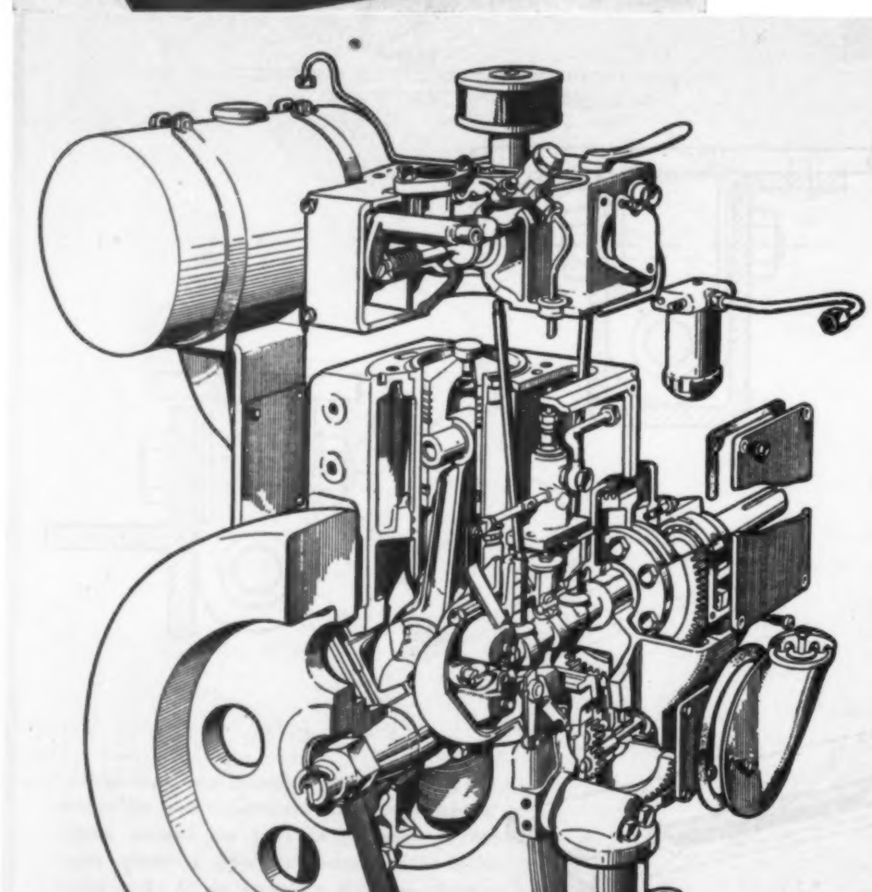






Small British Diesel Uses Unified Screw Threads

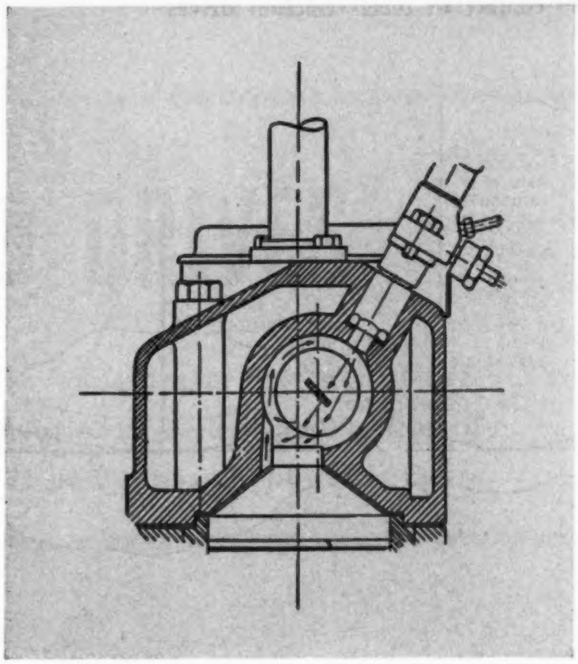
ALL threads in the new "Husky" diesel conform to the unified thread system adopted as standard in the United Kingdom, Canada and the United States. The small general-purpose diesel developed by Russell Newbery & Co. Ltd., Essex, England, incorporates a patented combustion system which provides easy starting and quiet running even on low-grade fuels. The vertical, single-cylinder engine, with a bore of 3¾ inches and a stroke of 4½ inches, provides 5 to 8 horsepower on a 12-hour rating basis over its speed range of 1000 to 1500 rpm. It is particularly suited to agricultural machinery and diesel-electric sets (as shown).



Inlet and exhaust valves are arranged horizontally and can be removed for servicing without removing the cylinder head. One main bearing housing is detachable to facilitate crankshaft removal, while the other is integral with the main crankcase casting. Cylinder liners are of the wet type. Piston is of low-expansion aluminum alloy; connecting rod is an H-section alloy steel forging with the big-end bearing split at an angle to simplify removal

through the cylinder bore.

All bearings and the valve gear are pressure-lubricated by a gear pump which delivers the oil through a "full-flow" filter; a pressure regulator and gage are included in the circuit. The horizontal, chilled cast-iron camshaft is driven from the crankshaft by a set of helical gears, of sufficient capacity to drive a half-speed power take off. Valves are operated by a conventional push-rod and rocker arm system.



Heart of the combustion system is the special design of the cylinder head. During the compression stroke the air charge is forced through the tangential throat and given a rapid swirl. At the moment of injection, the plug on the piston enters the throat and, in combination with the cut-away passage, creates definite, controlled turbulence which effects smooth and complete combustion without detonation. The absence of peak pressures is said to prolong life of the working parts greatly, while even heat flow from the apex of the piston eliminates sticking rings and causes minimum liner wear.

Wet Blast Cabinet Operates Without Pump

NO AGITATING pump is used in the Belmont wet-blast cabinet of Vacu-Blast Co. Agitation of the slurry, a water suspension of fine abrasive, is provided by an ejector, which draws liquid from the top surface and shoots it tangentially into the mixing chamber. An annular air jet in the blast gun provides strong suction to draw slurry to the gun, and accelerates the slurry to high speed through the converging nozzle. A high-pressure 700 cfm centrifugal blower pulls vapor from the cabinet through the filter. Water spray cleans the filter and window, and drains to a separate sump. High intensity flood lamps give a high level of illumination.

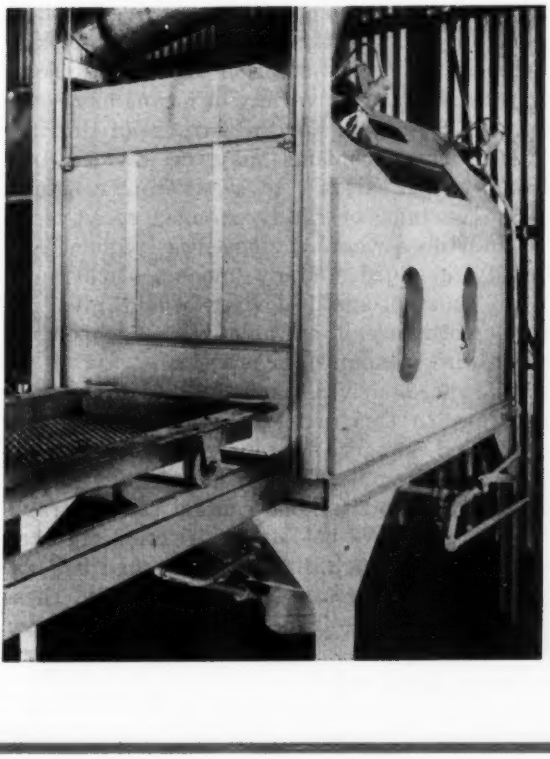
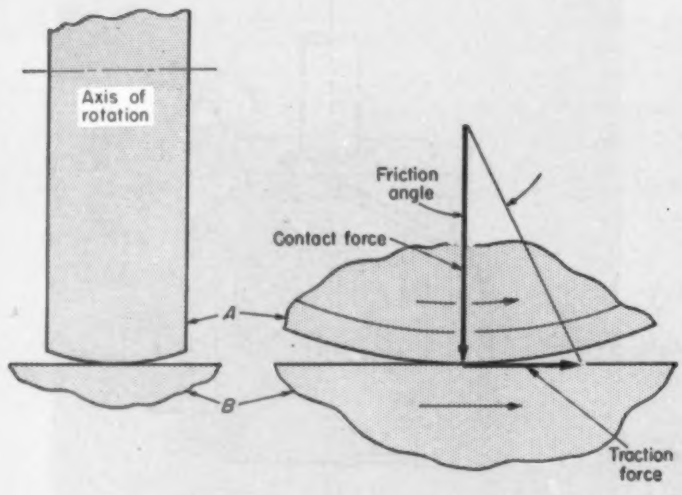


Fig. 1—Fundamental geometry of contact in metal traction drives



New Approaches

By Charles E. Kraus
President
Kraus Automatic Machines Corp.
Rochester, N. Y.

SOME form of drive means is required for every machine—machine being here defined in its broadest sense. Correct selection, design, and application of that drive are often fundamentals in the design of the machine and frequently determine whether the machine performs properly. The drive may take the form of a spring which drives your wristwatch, air or hydraulic cylinders, solenoids, weights, winds or tide, but by far the greatest numbers of drives involve rotational power applied to shafts.

The number and complexity of drive applications are as great as the number and complexity of machines, and in the solution of drive problems many types and a great variety of means have been utilized and developed. For the purpose of this article these drive problems may be divided into two broad classifications: those where a constant, or nearly constant, or rarely changed speed is required, and those where the speed is frequently or continually changed. Many machines currently built for constant-speed devices would give improved performance if suitable means for changing speed were economical or practical.

Variable-speed drives have been developed in many forms. In this article developments in new forms of speed changers are reported. These new arrangements all employ the principle of traction drive, relying upon frictional contact for the transmission of forces. All-belt drives, friction clutches, and contact between a tire and the road are examples of friction drives. More specifically, these new developments are in all-metal traction drives, commonly exemplified in principle by a steel locomotive wheel and a rail.

All-metal traction type speed changers to be discussed in this article have the additional characteristics that these surfaces are hard and highly finished and operate fully enclosed in the presence of low-viscosity, frequently special, oil. They could be considered as having evolved from earlier friction drive designs that utilize tarred fiber or leather disks but today they are as different as modern automatic transmissions are from those same friction drives applied to early automobiles.

All-metal traction drives offer several advantages: they are are easily adjustable, stepless throughout their speed range, compact, and require no auxiliary control equipment. Costwise, they are in line with other drive systems.

To their disadvantage in the past has been the handling of load surges and temporary overloads. Practically all other types of drives either have very large momentary overload capacity or are relatively unharmed by such overloads. Gear-driven units have as a reserve the breaking strength of the teeth, although normally operated well below such loads for long wear life. Most adaptations of electric motors have considerably higher stall torque than running torque, particularly dc motors. Overload conditions do no harm if overheating is controlled. The various belt drives can be damaged by excessive slippage but such slippage affects belt life only.

It is obvious that with all-metal traction drives there can be but two practical viewpoints on the problem of overload. First, the resulting slippage can cause distributed wear with no other major damage or, second, the design must be such that no slippage can occur.

to VARIABLE-SPEED DRIVES

. . . a report on fundamental research in metal-to-metal friction drive and application of the findings to new types of speed changers

How this problem has been approached will be discussed first. Then, how the resulting information has been applied successfully in the design of new types of drives will be described.

Nature of Contact: Certain basic fundamentals are present in all-metal traction contacts. *Fig.* 1 shows such a contact with the axis of rotation of rolling member *A* parallel to the contact surface and the motion direction of race member *B*. When the traction force is approximately zero, the conditions are the same as in a radial ball bearing under pure radial load. Contact force capacity is the same, as is the hardness effect of the members and the finish requirements. Under no load the contact is theoretically a point having no area. Under load, deflections occur in both members at the contact point, producing an area of contact with stresses varying from edge to center. This deflection is normally very small but the area may be appreciable, *Fig.* 2. The surface of race member *B* is depressed and rolling member *A* is flattened. Position of *A* relative to *B* is changed by the sum of these two deflections. For a given average life and set of conditions, the allowable contact force varies as the square of the curvative diameter. Ultimate failure in the race contact surfaces is due to fatigue caused by repeated stress reversals and normally occurs beneath the surfaces, resulting in a spalling or flaking. Surface wear, in the absence of dirt, is not present. Failure, when it occurs, is apt to be quite fast. Useful life is an inverse function of speed or a direct function of the number of stress reversals.

Next to be considered is what happens when *A* drives *B*. When two members are in contact and an attempt is made to move one relative to the other, a resisting tangential force immediately appears at the contact. This friction force is as fundamental and inevitable as deflection under load. The ratio of friction (or traction) force to contact force under conditions of slip or incipient slip, is known as the coefficient of friction and is the tangent of the friction angle, *Fig.* 1.

The coefficient of friction between two highly finished, hard metal surfaces is a function of the materials and the presence and nature of lubricants on the contact surfaces. Since all metal traction type speed changers are being analyzed here, it will be assumed that a lubricant is always present. Under such conditions the coefficient of

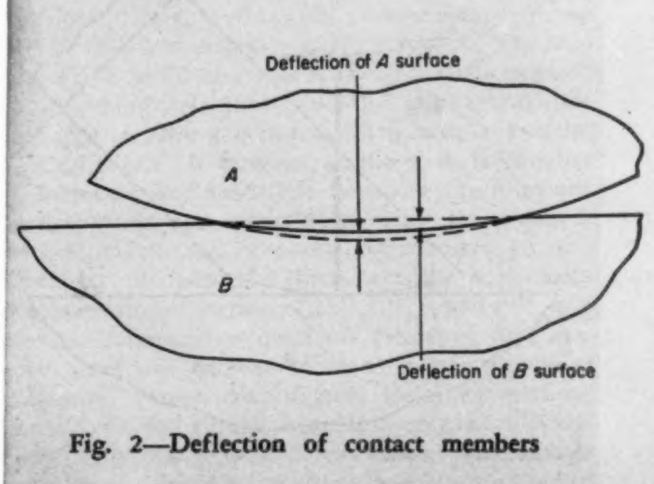


Fig. 2—Deflection of contact members

friction is found to be a function of speed and is greatly influenced by the viscosity of the lubricant. This is understandable since the contact is a rolling one and the lubricant must be squeezed from the contact area upon which rolling is to occur. How "dry" this contact is squeezed depends on speed of rolling and viscosity of the layer of lubricant. Time, although infinitesimal, is required to squeeze out this lubricant. Regardless of time, however, there is always some lubricant in the contact area, perhaps only in a layer of molecular thickness. As might be expected then, characteristics of the lubricant such as oiliness affect the coefficient of friction as well as viscosity.

During development of new designs of these speed changers, many tests have been run to determine coefficients of friction. Two types of test equipment were used. The first used adjustable means for presetting a known contact force and by means of prony brake and dynamometer equipment, different values of traction force up to the slip point were imposed. Fine-finish hard steel surfaces were used and various lubricants were tested at selected rolling speeds. The second type equipment used wedge members (sprag system) to preset the ratio of traction to contact forces and was loaded by dynamometer equipment. Tests again were run at different rolling speeds and with different lubricants. This second type equipment was especially valuable in determining wear and types of wear and in running accelerated life tests. Results are summarized in the following paragraphs.

At slow rolling speeds (about 200 fpm) the coefficient of friction was not greatly affected by variation in viscosity of lubricants up to 300 Saybolt seconds universal at 100 F. Values of from 0.08 to 0.09 were typical. As the rolling speed increased the coefficient dropped rapidly with the higher viscosity values, so much so, that no tests were made with lubricants above 100 SSU at 100 F at rolling speeds over 2000 fpm. At this speed typical coefficient values ranged from 0.06 to 0.07, and at rolling speeds of 4000 fpm the values dropped to 0.045 to 0.055. Difference in values between high oiliness (some compounded) and straight mineral lubricants was apparent and is reflected in the range of values noted.

These tests were correlated with the wear tests and it was concluded that the ideal lubricant should be very light, at least for high rolling speeds, in the range of 60 to 80 SSU at 100 F and should have good oiliness characteristics even though the coefficient of friction was somewhat lower. A completely neutral, noncorrosive and very stable lubricant is required. In some tests, serious corrosion was found in closely fitted surfaces subjected to intermittent stresses, as in fits between housing and races. Corrosion was not evident on the contact surfaces but may have contributed to wear.

Also in relation to the type of contact shown in *Fig.* 1, there may be some interest in observations on race failures and wear when the axis is parallel to the contact surface. When tests were run with a ratio of traction force to contact force well below the slip point, no surface wear was visible and failures were by subsurface fatigue. Just at the slip point, the first slippage seemed to be intermittent and above the slip point the surfaces were destroyed by seizure and galling. The point to note is that up to a quite definite slip point, no slippage nor indication of slippage in surface wear was found in parallel axis tests and, as may be expected, the allowable contact forces appeared to be similar to forces in bearing contacts where no traction force is involved. In other words, the use of the contact as a traction element, when the ratio of traction to contact force was safely below the slip point, does not appear to noticeably affect allowable contact load. It may be noted here also, that all tests to date, although not conclusive, indicate that this is true regardless of the type of contact. Factors that diminish capacity appear to affect both traction and nontraction contacts about equally, when all other conditions are the same.

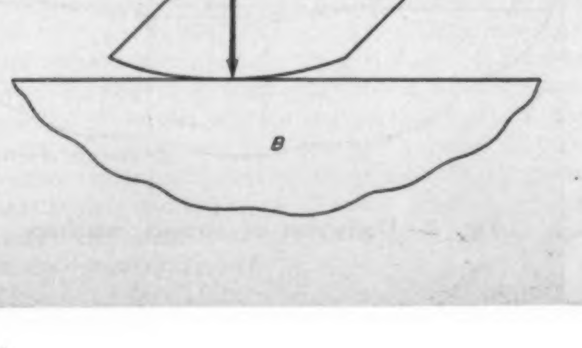


Fig. 3 — Left — Generalized contact condition used in most all-metal traction drives

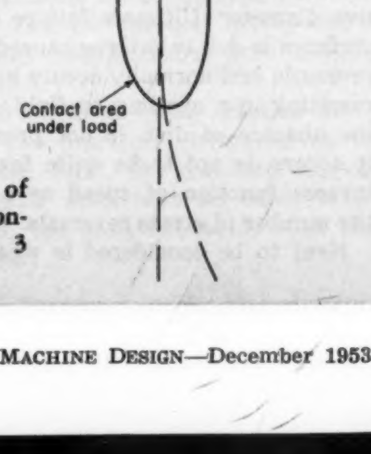


Fig. 4—Right—Paths of motions for contact condition shown in *Fig.* 3

Diagrammatically, *Fig.* 3 shows the more general type of contact found in traction type speed changers. The axis of the rolling member A is no longer parallel to the contact surface but at some angle to it. *Fig.* 4 indicates how the paths of motion of the two members, although tangential within the contact area, do not follow the same path. It is apparent that the tilting of the axis has introduced a new factor. Where first there was a pure rolling action, there is now a spin action within the heavily loaded contact area. Amount and seriousness of this spin action are dependent on the inclination of the axis, the relative diameters, and the size and stress distribution of the contact area. This same spin action is found in many ball bearings and is quite pronounced in thrust bearings operating at speeds that produce considerable centrifugal force on the balls. As would be expected, this action introduced surface wear which was not noticeably present with the axis parallel. Although, to date, tests again are not conclusive, it appears that this spin does not greatly affect the coefficient of friction. However, the oiliness characteristic of the lubricant does definitely have an effect on the rate of wear. With better oiliness, a small drop in coefficient of friction results in a small increase in necessary contact force for a given traction force and offsets to some degree the reduction in wear due to the increased oiliness. Nevertheless, and paradoxical as it may seem, the conclusion is that for best overall results these all-metal speed changers operate best, life factors considered, with a friction decreasing lubricant on the contacts that operate the units through friction.

Determining allowable contact forces for a practical speed changer life is complicated by many factors. The contact area for a given load is a function not only of the curvature diameter but also sometimes of the restriction in the width of the area by the geometry of the parts. Wear, rather than fatigue, determines average contact pressure and contact forces; therefore, these factors must be lower than would be indicated by antifriction bearing type computations. Wear, in turn, is a direct function of the oiliness of the lubricant and of the frequency of contacts over any given surface. For a given design, since many of the factors involved may be peculiar to the design, the final determination of practical ratings must be obtained by life or field tests. Closely simulated laboratory tests are, however, of great value in determining design assumptions and in the selection of lubricants and construction materials.

What then are the possibilities of all-metal traction type speed changers? Past experience with them shows that they are compact devices, self-contained with auxiliary controls unnecessary, that they can have practical ranges of output speed, that this speed is easily adjustable and completely stepless. It is known that they must be rated on the basis of wear rather than fatigue but, from field experience with existing production designs, it is also known that the capacity is high, ranging from 1 to over 3 hp per contact. Furthermore they perform best with special lubricants and cleanliness of the lubricant must be safeguarded. Past experience has also shown that difficulties, other than those caused by improper or contaminated lubricant, have been caused by overload applications, continued operation for long periods at a fixed ratio causing localized wear, and inability of some designs to take rapidly applied or fluctuating loads without slip. Efficiency, not yet mentioned, varies with current production models from zero at zero output speed, an extreme obtained in conjunction with planetary gearing, to values approaching 90 per cent with favorable ratios, obtained even though the designs contain bearings that are subject to contact force loads.

From laboratory tests, from work on new designs and from tests on pilot models of these designs, it has been concluded that past difficulties can be eliminated, that capacities can be further improved and that overall efficiency can be consistently high. Difficulties due to overloaded applications can be overcome by proper rating or by education of the user in the nature of the mechanism he is using. Difficulty with localized wear can be minimized by periodic varying of ratio setting to distribute the wear over greater areas. This practice is almost always possible and again requires understanding by the user of the nature of the mechanism. Overload and fluctuating load can be handled by proper design. Here it is essential to have the contact force build up with load practically instantaneously and requires attention to the inertia of the parts affected and to the structural deflections accompanying the build-up that operate to limit its magnitude. Consistent high efficiency requires, of course, the elimination of sources of energy loss. Losses at the contacts are inherent, but bearings subjected to contact forces are loss sources that can be eliminated.

Theoretically it appears possible to have operating characteristics similar to gear drives in these units. A gear has, as a reserve, the breaking strength of the teeth and, except when exactly on pitch line, is subject to a scuffing action. Wear on the teeth is a function of lubricant and load. For long life the allowable normal loads are appreciably below the structural strength. The traction drive units have, as a reserve, their capacity to withstand brinelling, and the spin action present on the rolling contacts introduces a scuffing action. Wear on contact surfaces is a function of lubricant and load. For long life the allowable normal loads are appreciably below the compressive strength. A necessary requirement of this similarity of operating characteristics is a design that permits utilization of this full reserve of compressive strength on overload. Bearings that support gears are subject to loads not in excess of tangential forces transmitted. Bearings in traction drives, for similar bearing losses, should also carry only tangential force loads, not contact force loads. From a practical standpoint, gearing should be designed with strength sufficient to stall

their power source and this strength requirement should normally be less than their breaking strength to allow an operating factor of safety. Traction drives also should be designed to be able to stall their power source without damage with an operating factor of safety and this implies not only that no brinelling occurs but also that no slipping condition develops.

In the remainder of this article new developments in traction drives will be discussed and a number of designs that meet all of these requirements will be presented.

Dual Traction-Ring Design: *Fig.* 5 shows a pilot model of one new development in all-metal traction type speed changers. A schematic view of the traction elements is shown in *Fig.* 6, which also shows a gear drive from the speed-changer jackshaft back to the input drive centerline. This unit can be designed for output speed ratios up to 10 or 12-to-1 and this range for a given input speed can be easily positioned by selection of the output spur gearing. The basic traction elements are a double conical spool-section disk, normally used as the drive member, keyed to the input shaft; output disks consisting of two matching conical surfaced members rigidly attached to a heavy shaft section so as to become essentially a single spool-shape member; and two ring-shape traction members interposed between matching conical surfaces of input and output disks.

The entire mechanism is enclosed in an oiltight housing, the input and output shafts are suitably journaled and sealed at the housing and the jackshaft is journaled in a carrier (not shown), which maintains its axis parallel with the axis of the input shaft. The traction rings are carried in a cage and supported and positioned by rollers (also not shown). The drive ratio between the input shaft and jackshaft is determined by the position of the rings, as is obvious from *Fig.* 6. Adjustment of ratio is accomplished by tilting the ring support cage which causes a self-energized progression of the rings to the desired ratio producing location, without introducing a sliding or scuffing action.

An enlarged section at the contact points is shown in *Fig.* 7. The sides of the traction rings are ground to a large radius and finely finished. They contact the finely finished conical sides of the input and output disks. All three parts, of course, are hard. This combination of three elements is similar to the combination of a ball in contact with two races, with one essential difference. The radius of the contact surfaces is considerably larger than half the width of the ring. By this means, the plane of the ring is stabilized parallel to the planes of the contact surfaces. The contacts are maintained directly opposite so the contact forces are in line.

Fig. 8 shows an enlarged section of these contact points in a plane perpendicular to *Fig.* 7 under conditions of no traction load. Contacts are opposite each other and are in the plane of the axes of the disks. When no tractive effort is present, contact forces are only sufficient to insure

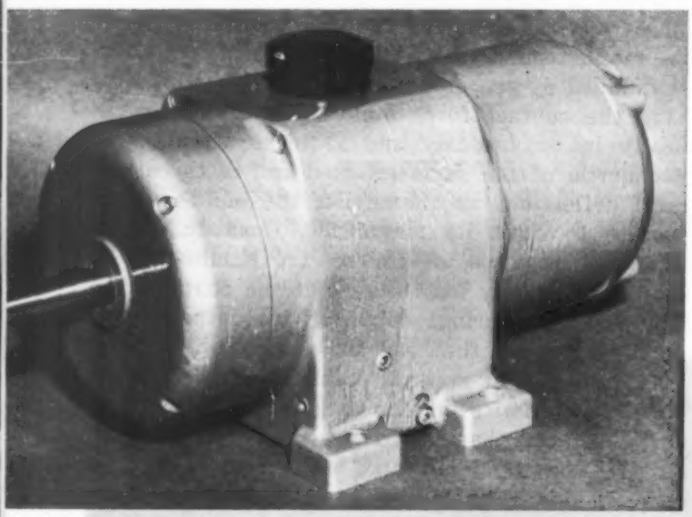


Fig. 5—Above—Encased unit of the dual traction-ring design

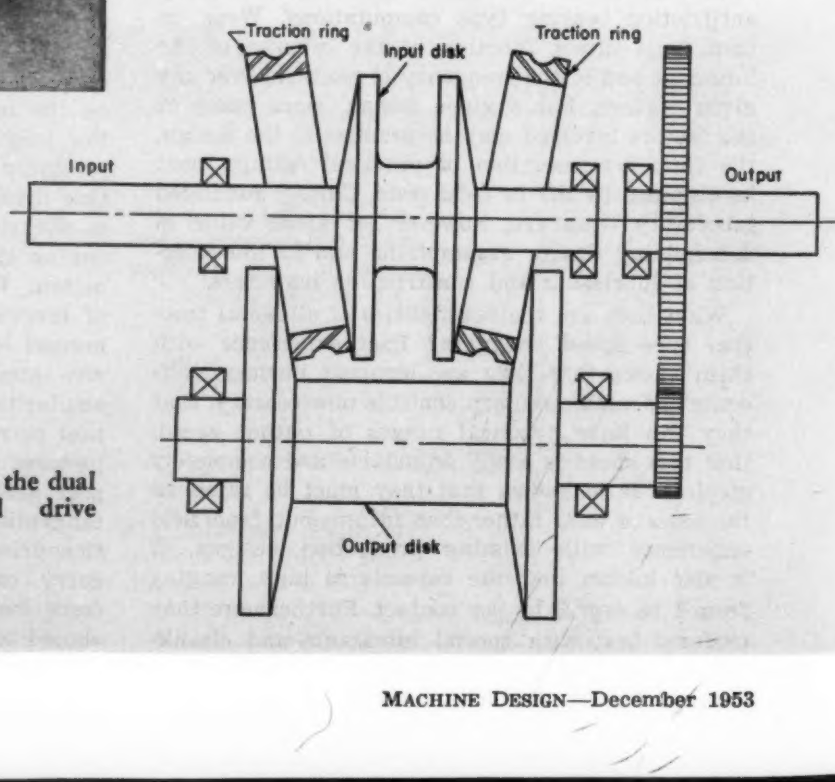


Fig. 6—Right—Schematic of the dual traction-ring variable-speed drive

actual contact. Such contact is maintained by an automatic take-up means (not shown) which imposes a small preload and serves also to take up normal small increments of play which may be present in the bearings.

If a load (torsional resistance) is put on the jackshaft (drive output disks by input disks through traction rings), traction forces are produced at the contacts in the ring members as labeled in *Fig.* 9. Note that these traction forces are in opposite directions on the two contacts and impose a force couple on the rings. If all members were completely rigid, the rings could not swivel in the plane of *Figs.* 8 and 9 and the resulting contact forces would be infinite.

Therefore, the amount of ring swivel and the ratio of traction to contact forces are functions of the deflection of the elements. It is apparent that in swiveling the ring, the input and output disks are wedged apart at the contact points and reaction forces from the contacts are in line. This swivel action induced in the rings as a function of traction effort and controlled by deflections in the component members is a basic fundamental of this design. Given the maximum traction force the design is to handle, adequate to stall the power source, and knowing the maximum ratio of traction to contact force that is safe for the operating speeds and lubricant, one can calculate the width of the rings and the required sum of the deflections of the driving and driven members. Compression deflections at the contact areas are very small but can be taken into consideration. Since the required deflections are a function of the ring widths and since the traction to contact force ratio decreases with increased ring width for a given angle of swivel, it is obvious that relatively wide rings are essential for practical design.

Referring to *Fig.* 6, it can be seen that contact forces on the input disks are balanced within this member and, except for a small component due to this angle of the conical surfaces, these forces are not transferred to the journal bearings. The conical angle is relatively small, not only to keep this component force small, but also to increase the effective curvature diameter of the contacts with accompanying increase in load carrying ability. Deflection at the input contacts is equal to a component of the input shaft deflection plus the deflection of the flanged conical sides. It can be seen that this latter deflection is a function of the radial position of the contacts and is about zero near the shaft and a maximum at the peripheral point. In a like manner, the total deflection at the output contacts equals the shaft and the disk deflections. These deflections are a minimum when the contacts are at the periphery of the input members. Contact forces again are contained within the rotating assembly. Bearing deflections do not enter into either total to an appreciable extent because such loading as they carry is purely radial, and initial play, if any, is kept out.

Since total deflections of all members for any contact position can be calculated, it is possible to proportion or design shaft sizes and all drive members. The resulting shape of these members is about as shown and the flanging of the input member and the use of the relatively heavy shaft diameter of the output disks are necessary and are dictated by requirements.

Ability to handle suddenly applied overload or fluctuating loads is determined by the ability to buildup contact forces to the extent required and fast enough to match the load. With this design the action is almost instantaneous. The only rela-

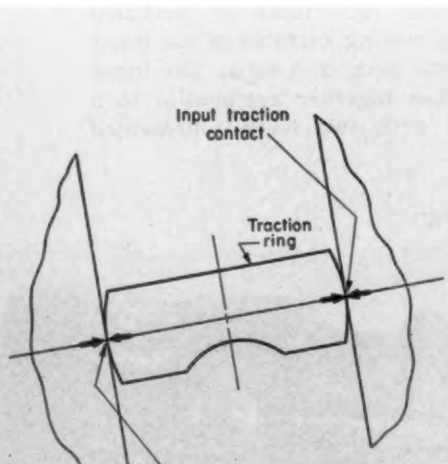


Fig. 7 — Section detail showing contacts on the traction ring

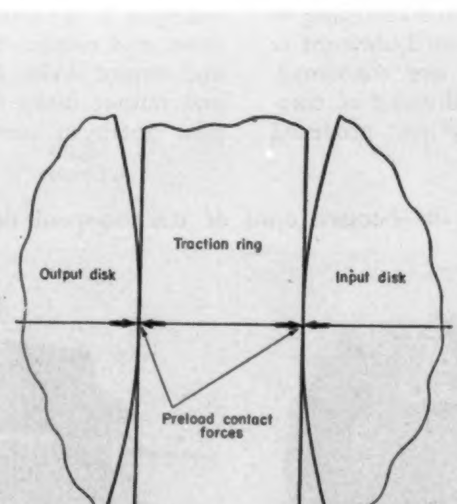


Fig. 8—Section detail showing contacts on the traction ring, under no load, in a plane perpendicular to *Fig.* 7

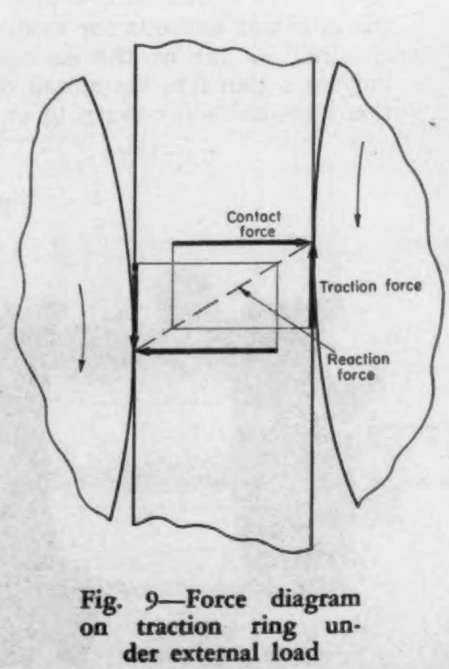


Fig. 9—Force diagram on traction ring under external load

tive motion involved is a very small tilt of the two low-inertia traction rings.

Buildup of forces is not disturbed by deflections of either component members or bearings. This latter source of deflection may be a serious one, particularly if radial thrust type bearings are involved carrying contact rather than traction loads. For high efficiency, bearing losses are a minimum and all heavy contact forces are contained within the rotating parts. From cost and maintenance standpoints, the basic elements are simple and rugged. Wear, which is inevitable with all designs, can be held to reasonable limits by proper rating of the units, proper materials and good workmanship.

This design makes use of two input and two output traction contacts. As presented, it is particularly suited for the range of about ¼ hp to 5 hp and output ratios from about 4-to-1 to about 12-to-1.

Traction-Spool Designs: Two important factors to consider in the design of all-metal traction-type speed changers are the rate of wear on contact surfaces, and the efficiency. The rate of wear, assuming clean lubricant, is influenced by the area of the contact and by the effect of the spin action always present. The area of contact, in turn, is a function of the curvature diameter, the hardness of the surfaces and the magnitude of the contact forces. Wear does not appear to be noticeably affected by the presence or absence of traction forces if these forces are safely below the slip point. Both laboratory tests and field experiences have shown that, when contact forces are relatively low, wear becomes negligible.

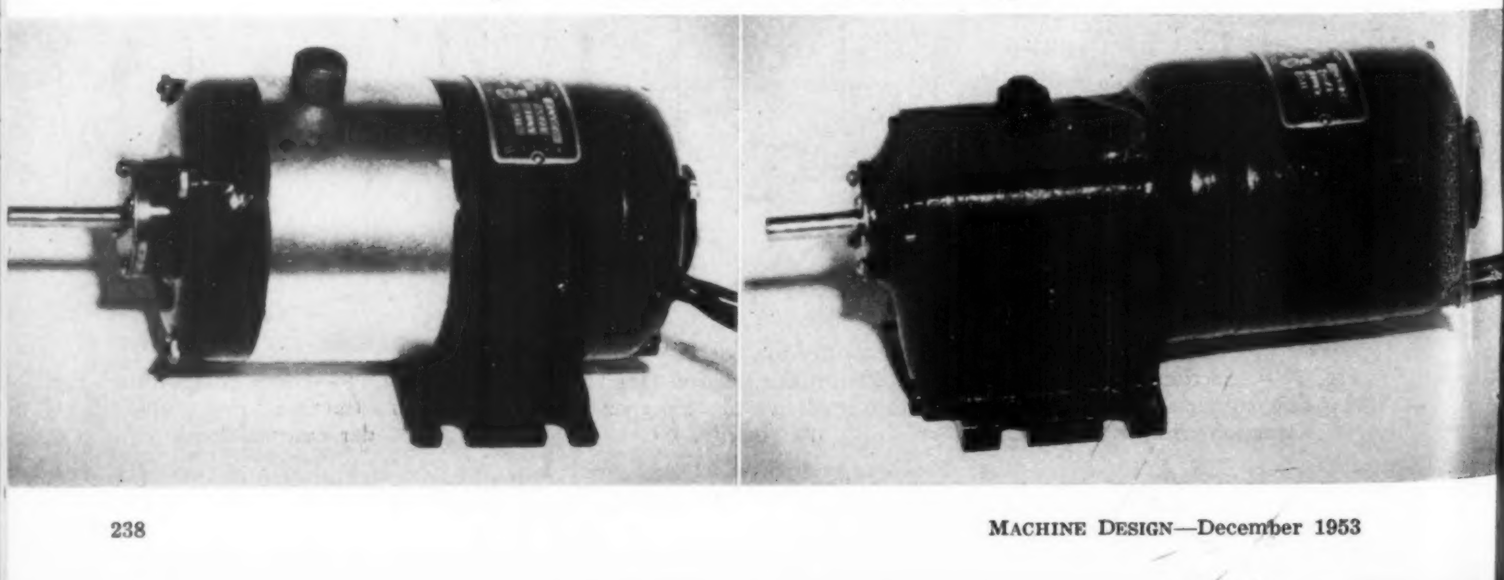
Efficiency is affected by energy losses caused primarily by three sources. Bearings supporting the various elements are an important source of loss only if loaded by contact rather than traction loads. Under traction loads, well designed bearings should not account for over 1 or 2 per cent of the input power at full load. Windage and churning of the lubricant accounts for more loss. Lubricant is required, as far as the contacts are concerned, only as a thin film but splash or dipping of traction elements is necessary to carry heat produced to the housing walls for dissipation. If the lubricant level is maintained too high this source of loss can be appreciable, the same as sometimes is found in overlubricated high-speed gear boxes, but a lubricant level sufficient for the requirements does not cause much energy loss. Losses from this source seldom exceed 1 or 2 per cent.

The third source of energy loss occurs in the contact areas and is caused almost entirely by the spin action. With low contact forces, the contact areas are small and these losses are low. Actual figures are dependent on too many factors to make possible any but the most general indication of range, but in percentage of energy input such losses range from 3 or 4 per cent on very light loads to 10 or 15 per cent on loads approaching the brinelling point of the races. The range indicated is, of course, predicated on the use of selected lubricants, fine finished hard surfaces and the absence of abnormal conditions such as misaligned parts or dirt.

If a speed changer is to be designed for use only with light loads, as would be the case for subfractional powers of, say, 1/6 hp maximum, advantage can be taken of the low wear rate and energy loss of light contact loads and a very simple mechanism can be produced by allowing a maintained maximum contact force. The traction-spool design is of this type.

Shown in *Fig.* 10 is a photograph of a pilot model of such a unit and *Figs.* 11 and 12 show the schematic arrangement of the simple traction elements used. Input and output shafts carry disks having on adjacent faces suitable gooves for balls to carry thrust between the disks. The nonadjacent faces have as a radial profile a gear tooth form, as for example an involute or modified involute. Supported on a suitable axis, carried in the present instance by a U-shape bracket, is a spool-shape third member. The inner surfaces of this member also have as a profile a gear tooth form, as for example a straight-side rack tooth or modified form, and contact the mating surfaces of the input and output disks. Note that, in a sense, the input and output disks taken together are similar to a gear tooth in mesh with two teeth represented

Fig. 10—Encased units of traction-spool design



by the flanges of the intermediate spool. By tilting the axis of the spool in the plane of *Fig.* 11, the points of contact on the input and output disks move radially in opposite directions and wide ratio range is readily obtainable above and below input speed. Tilting of the spool is accomplished by a small rotary movement of a lever to which the U-shape bracket is pivotably attached. At either side of the central position, this bracket rocks away from the centerline of the units by a small amount dependent on the profile shapes and "tooth" width. A spring centrally attached to the bracket is used to furnish operating contact forces adequate for the power to be handled.

The entire mechanism is enclosed in an oiltight case and the input and output shafts have suitable bearings and seals. The adjusting lever it attached to an inside wall of the case by a stud shaft. Aside from the requirement that input and output shafts be on the same centerline for proper alignment of the ball thrust bearing separating the disks, there are no tolerance or alignment problems. Contact surfaces must have correct profiles and fine finish, but localized wear, if it should occur, cannot affect ease of ratio adjustment. There is one drive contact on input and one on output. Theoretically capable of handling well over one horsepower, the design should be limited to subfractional power for reasons already outlined.

Although the ball thrust bearing carries contact forces, these forces are low. Moreover, since both input and output shafts rotate in the same direction the thrust bearing must accommodate only the difference in their speeds. Except for this bearing, all contact forces are contained in the rotating members and all other bearings handle mainly traction forces. Total ratio range can be limited by design and by introducing stops on the tilt-control lever to limit the adjustment. Reductions to 5-to-1 and speedups of 2 or 3-to-1 have proved practical from pilot model testing. The unit is compact, reversible, inexpensive to manufacture and almost completely free from maintenance. It is presented for use in the countless applications where a low-cost, trouble-free speed changer is advantageous for subfractional power.

High-Reduction Planet Type: The two previous new developments have been concerned with designs which produced satisfactory ratio ranges above and below input speed. The position of this range, of course, can be altered by gearing. Any ratio of gearing can be used, but the ratio of maximum to minimum speed remains unchanged.

Before a discussion of the development of the high-reduction drive, some comments appear to be in order concerning the combination of a speed changer and planetary gearing. Simple planetary gearing is composed of a sun gear, a ring gear and a planet carrier with one or more planet gears. Normally one is the drive member, one is driven and the third is locked stationary. However, two members can be drivers by different sources and the third can be the driven member, with no member locked. If the ring and sun members are rotated in opposite directions or the sun and the carrier in the same direction, their speeds can be varied relative to each other, and the speed of the driven member can be varied down to zero speed or either side of zero. With traction type speed changers, there are always two rotational speeds available: the constant input speed obtained from the motor or drive source, and the adjustable output speed.

With the traction ring design, such planetary

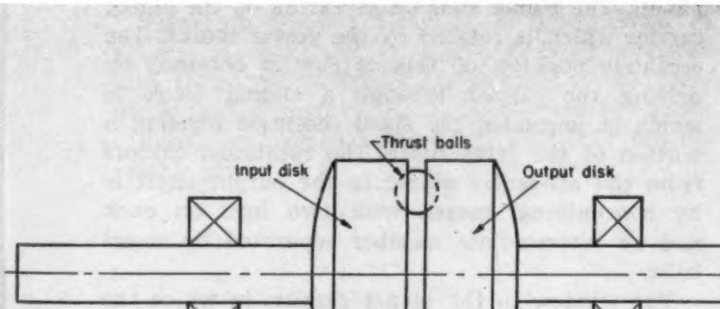


Fig. 11—Above—Schematic of traction-spool variable-speed drive

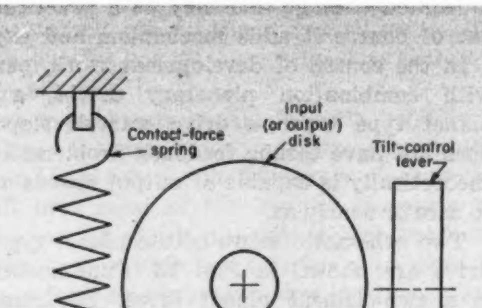


Fig. 12—End view of traction-spool design

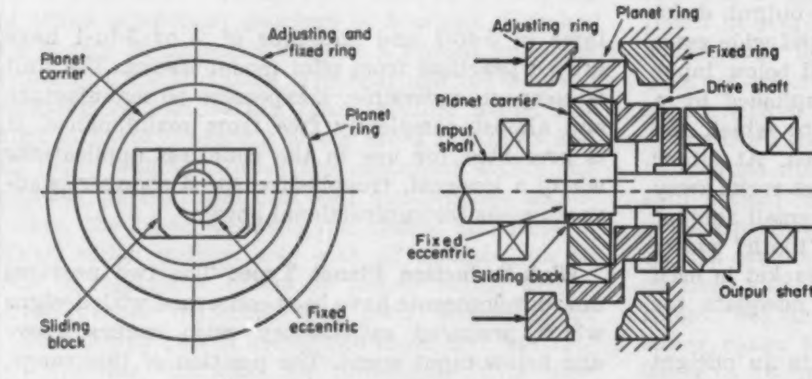


Fig. 13 — High-reduction planet type variable-speed drive

take-off gearing can be used very simply since the sun gear can be attached to an extension of the drive shaft, and the jackshaft can drive the planet carrier. In this case, both drive in the same direction but the sun gear can be rotated very rapidly relative to the adjustable speed of the planet carrier, and output speeds to zero are easily obtained. Studies have been made of the gear loads and of the division of power between drive sources. Pilot models of such gearing, in combination with the traction ring design, have been tested.

A major problem arises as the output speed is lowered toward zero, stemming from the fact that the speed changer has constant power characteristics. The planetary gearing, below certain output speeds, introduces a feedback of torque to the traction contacts which can become destructive at or near zero output speeds. Such combination drives can be rated for constant or tapered torque, but still dangerous contact forces could result from abnormal load. One pilot model incorporated a slip clutch on the output shaft as a safety feature. This prevented damage and may be a practical answer, but of course it adds mechanism and expense.

In the course of development work, particularly with combination planetary drives, a straight planet type traction drive was developed which does not have torque feedback problems but which theoretically is capable of output speeds adjustable to zero or nearly so.

Two schematic views of the planet type traction drive are shown in *Fig.* 13. Fundamentally, this is a two-element planet drive. The ring gear is stationary and a single large planet is carried on a planet carrier which is the drive element. The single planet gear is the driven element, and output power is taken off by an all-metal connection from planet to output shaft which compensates for the off-center position of the planet axis. It is obvious that, for each revolution of the planet carrier, the point of contact between the planet and the stationary ring also makes one revolution and, therefore, the planet rotates backward by the distance that the circumference of the ring exceeds the circumference of the planet. If an adjustable pitch is introduced, it can be seen that the rate of rotation of the planet about its axis approaches zero as the diameter of the ring approaches the diameter of the planet. This type of planetary, using special forms of gears for close matching of diameters, is well known but it has never been successfully applied to an adjustable traction type drive.

The design shown in *Fig.* 13 is the result of applying the results of many tests and the tedious evolution of certain definite requirements that have dictated specific geometry of members. Two rings are keyed to the case, one movable axially by an adjusting means (not shown), and the other fixed. The planet ring rolls on these two rings and has one contact point on each ring. The adjacent surfaces of the stationary rings have a special formed surface to be discussed in a brief mathematical analysis to follow. It will be seen that as the adjusting ring is adjusted toward or away from the fixed ring that the contact points are moved closer or further from the center axis, effectively changing the pitch circle of the rings and consequently changing the rotational speed of the planet. The planet runs on a bearing on the planet carrier which is rotated by the power source. The eccentric position of this carrier is obtained by driving the carrier through a sliding block in which is journaled the fixed eccentric forming a portion of the drive shaft. The rotational takeoff from the off-center planet to the output shaft is by conventional means with two lugs on each and an intermediate member separated by caged balls.

The window in the planet carrier, in which the sliding block is gibbed, is off center by the amount of eccentricity of the eccentric so that the planet can come to a concentric position relative to the axis of the rings and the input and output shafts. Stops in the adjustment means can prevent movement to this extreme position. Although this fundamental arrangement ideally answers the requirements of the drive, the block design can be changed to a connecting rod design without greatly altering the proper functioning. The drive shaft eccentric cannot, however, be directly journaled in an off-center location in the carrier. This latter construction can be shown mathematically to be completely inoperative.

Since these traction type speed changers are essentially constant-power devices,

$$P = N_o T_o$$

where P = available power, N_o = output speed, and T_o = output torque. Also it is known that

$$N_o = N_i \frac{R_r - R_p}{R_p}$$

where N_i = input speed, R_r = radius of ring and R_p = radius of planet. Combining these two equations gives

$$T_o = \frac{PR_p}{N_i(R_r - R_p)}$$

Power P, input speed N_i and planet radius R_p can all be considered as constants, or $PR_p/N_i = k$. Term $R_r - R_p$ is recognized as the distance between the planet axis and the principal axis at any given time. The output torque therefore would vary as the inverse of $R_r - R_p$.

In the force diagram of *Fig.* 14, the eccentric has moved relative to the carrier, through an angle θ. Such movement has displaced the carrier vertically in the figure by distance $R_r - R_p$. It is obvious that

$$R_r - R_p = r - r \cos \theta$$

Therefore, the following relationship should hold:

$$T_o = k \frac{1}{R_r - R_p} = k \frac{1}{r - r \cos \theta}$$

Resolving moments about the main axis gives

$$F_i r = F_t (r - r \cos \theta)$$

or

$$F_t = \frac{F_i r}{r - r \cos \theta}$$

Therefore traction force F_t is proportional to $1/(r - r \cos \theta)$ and, of course, F_t is proportional also to output torque T_o. Accordingly, the foregoing proportionality, $T_o = k/(r - r \cos \theta)$, holds.

Resolving moments about the center of the eccentric gives

$$F_t r = F_r r \sin \theta$$

or

$$\frac{F_t}{F_r} = \sin \theta$$

Ideally the ratio of traction force F_t to contact force F_c should equal a factor, designated μ, for safe operation. That is,

$$\frac{F_t}{F_c} = \mu$$

and

$$F_t = \mu F_c$$

Substituting this equation in the foregoing moment equation gives

$$\frac{\mu F_c}{F_r} = \sin \theta$$

and

$$F_c = \frac{F_r \sin \theta}{\mu}$$

Since F_c is perpendicular to the contact surface, the angle of this surface from the radial should be the angle whose sine is $\mu/\sin \theta$. Therefore the form of the adjacent surfaces of the rings can be determined. It may be interesting at this point to comment on what would result if these surfaces were made, for example, conical or with constant angles of inclination. At some one point they could have just the right slope. As the contact points moved further out radially, insufficient contact forces would cause complete slip, or as θ approached 180 degrees, or zero, the contact forces could become excessive.

The foregoing analysis shows that the described mechanism can be designed, theoretically, to have the exact ideal requirements for constant-horsepower operation. Output torque can be made available in the exact amount required by the ratio of the input and output speeds, disregarding losses. Also, the contact surface wedge angles can be designed, by proper slope at the contact points corresponding to all output speeds, to produce just the value of contact force required for safe operation from the amount of radial force available. For example, if it is assumed that $\theta = 90$ degrees, sin $\theta = 1$ and the sine of the contact surface angle should equal μ at that point.

Reducing these findings to a practical design means in this case that the end of the range where θ approaches 0 must be modified inasmuch as the ring form goes to 90 degrees and then becomes indeterminate; the forces become very high and the adjustment difficult. It is necessary therefore to limit the maximum reduction so that output speeds do not go closer to zero than can be safely handled.

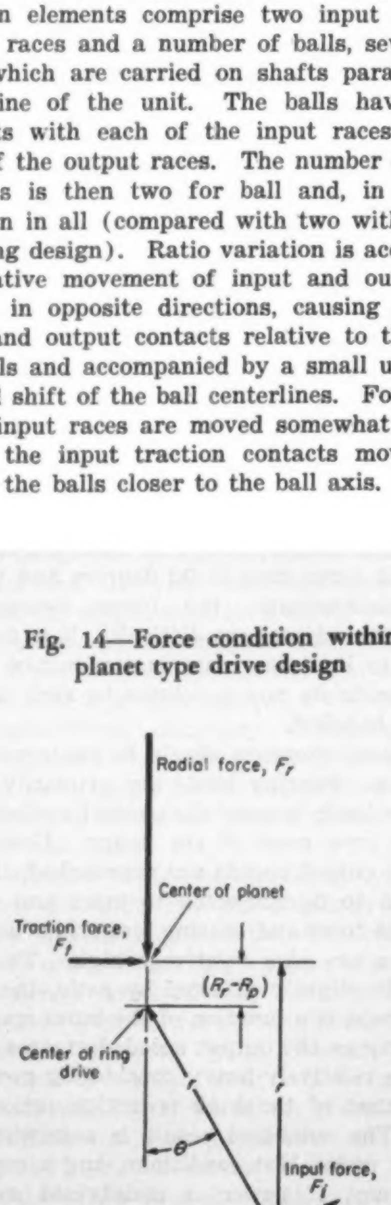
Two other comments should be made relative to this design. Bearing loads are primarily subject to traction loads because the contact surface angles are small over most of the range. However, as the lowest output speeds are approached, the bearings begin to be subjected to more and more of the contact force and bearing losses increase. Contact losses are also relatively high. This is because, only slightly affected by ratio, the contact moving speed is a function of the input speed. Also particularly as the output speed decreases, contact forces are relatively heavy considering power handled, because of the high reduction ratios of the design. The combined result is somewhat lower efficiency under best conditions, and a rapid drop in efficiency, if power is maintained as output speeds approach zero.

The design is reversible and is presented as particularly applicable for reductions from about 15-to-1 to about 350-to-1 and for a power range to about ½ hp.

Torque-Responsive Transmission: So far discussed have been three new developments in traction type speed changers, all designed for relatively low power ranges. The traction-ring design handles the most power of the three and it is limited to about 5 hp. None of these designs had more than two traction contacts to divide the input or output loads. Now about to be discussed is a radically different type design which will handle much higher power. This design is more than an adjustable source of output speed; it is actually an automatic torque-responsive transmission capable of performance not possible with any existing design and having ideal requirements for application to widely fluctuating loads and speeds.

Fig. 15 shows a cross section through the centerline of a unit incorporating the principles of this new development. *Fig.* 16 shows a view through the indicated vertical centerline of *Fig.* 15. The traction elements comprise two input races, two output races and a number of balls, seven in this case, which are carried on shafts parallel to the centerline of the unit. The balls have traction contacts with each of the input races and with each of the output races. The number of traction contacts is then two for ball and, in this case, fourteen in all (compared with two with the traction ring design). Ratio variation is accomplished by relative movement of input and output races axially in opposite directions, causing a shift in input and output contacts relative to the axis of the balls and accompanied by a small up or down vertical shift of the ball centerlines. For example, if the input races are moved somewhat closer together the input traction contacts move up the side of the balls closer to the ball axis. The balls

Fig. 14—Force condition within planet type drive design



are also moved radially outward a small distance and the outer races move apart, shifting the output traction contacts further out on the balls away from the ball axis. The reduced input moment arm and increased output moment arm acting on the balls therefore increase the relative rotational speed of the output races. A reversed movement of the races would decrease the relative rotational speed.

Essentially this traction type is equivalent to two-stage gearing. The input races and the balls at contact diameters corresponding to a given position constitute the first stage, and in a similar manner, the balls and output races constitute a second stage. For any relative position of the elements a definite overall ratio is fixed. Readily obtainable ratios can vary steplessly from about a 4-to-1 reduction to about a 2-to-1 speed up, for a total range of 8-to-1. The effect of ratio range on ratings of the unit will be discussed. It may be pertinent to point out, however, that while the general arrangement of these elements is similar to planetary type gearing, it is usually not so considered because, when the planet carrier is fixed, planetary type action is not involved.

The ball traction elements in this development are free to float axially and radially without restraint but circumferential movement is prevented. The parallel positioned ball axles are carried in blocks free to swivel on pins attached to nonrotatable parts fastened to the case. The ball axles are therefore subject only to reaction forces (the sum of the traction forces on each ball) and heavy contact forces are balanced within the ball element. Bearings are thin bronze, running on hardened steel axles and pressure lubricated by oil holes in the axles communicating through the drilled axles, swivel blocks and pins to a built-in pump. Pressure-velocity values on these bearings are constant regardless of ratio setting for a given input speed. Bearing clearances are relatively open to allow a reasonable volume of oil to be thrown off of the ends of the bearings as a spray. This oil lubricates the races and sprag systems and is thrown out of the mechanism through spaced drain holes and returned to a sump (not shown) for recirculation. It therefore is used to carry frictional heat from bearings and contacts to the case for dissipation as well as for lubrication. The traction elements do not run in a lubricant but rather in a heavy spray.

The output races transmit traction forces to a cylindrical housing attached to the output shaft, through a sprag system for each race. The sprag systems operate in a conventional manner and, in this case, comprise rollers adjacent to cam rings on one side, and the ground sides of the output races on the other side. A relatively large number of rollers are used and the cam shapes are specially designed as dictated by functional requirements to be discussed. The output races are not otherwise keyed or attached to the housing and are therefore free to move circumferentially relative to this housing a considerable distance with accompanying axial motion relative to each other

under the control of the sprag systems. They are keyed to each other by floating keys to insure that they move together circumferentially.

If the distance the output races can move is stopped at any point by external means it is clear that further ratio change cannot take place in the stopped direction. Since this unit is torque responsive, increase of output torque over that available at a given ratio from the preset input torque, will cause the ratio to increase and the output shaft to slow down. If the output load decreases, the output speed will increase. Therefore the race movement stop mechanism is designed to prevent an increase of output speed above the setting desired. In other words, if an output of 1000 rpm is desired the adjustable stop is set for 1000 rpm. Any output torque exceeding the torque available at that speed from the preset input torque value will cause a decrease in speed from 1000 rpm setting, but a lighter load willl not allow a higher speed. The stop mechanism shown takes the form of a band keyed for axial movement on the housing with wide helical grooves on the inside surface which act as stops for rollers carried on the keys. One side of these helical grooves stops the output race movement relative to the housing at a position depending on where the band is axially located relative to the housing. This position is determined by a roller, running in an annular groove on the outside of the band, which is attached to a journaled disk that is adjustable by an external knob.

Input races are also free to move circumferentially relative to the hollow shaft member driven by the power source, under the control of sprag systems, one for each race. These sprag systems are similar in design to those on the output races but are mounted opposite to conventional use. Instead of the hollow drive shaft driving the input races through the sprag systems, forces are actually fed back from the races through these sprag systems to the drive. It is by means of this recirculation or feedback of forces, that the torque-responsive characteristics are obtained. Input races are not actually driven by the input drive shaft but by an auxiliary shaft through a double key which extends on each side of the auxiliary shaft through windows in the hollow shaft. The auxiliary shaft is attached to piston elements in a rotary hydraulic cushion mechanism contained in a suitable case forming a part of the hollow shaft. Hydraulic pressure, controllable externally by a relief valve, is obtained from the same built-in pump used as a source of bearing lubricant and is used to transmit torque from the hollow drive shaft to the auxiliary shaft. Circumferential movement of the piston elements (and consequently the auxiliary shaft) relative to the cushion case is the same as that of the input races relative to the hollow drive shaft. It follows that the torque transmitted to the auxiliary shaft is controlled and determined by the lubricant pressure in the hydraulic cushion and, if the torque is increased or decreased on the auxiliary shaft above or below the amount that can be supported by this pressure, lubricant will either be bypassed or added to the cushion chambers. This in turn causes movement of the auxiliary shaft and input races relative to the hollow drive shaft. The input races therefore are moved, under control of their sprag systems, relative to each other. The resulting change in the ratio of the unit then decreases or increases the torque on the auxiliary shaft accordingly, restoring the torque to the amount that is supportable as set by the fluid pressure.

The net result is that a change in loading of the output shaft will cause, automatically, a change in the ratio setting of the unit since the change in output torque is transmitted back to the input races. It was mentioned that there is a force feedback through the input race sprags to the drive. This force is about equal to the traction force and means that the torque applied to the auxiliary shaft is about double that required for the traction forces on the ball contacts. The force from the sprag cams feeds back to the cushion case and through the fluid to the auxiliary shaft thereby completing the circle. Fluid pressure is consequently about double what it would be without this feedback and sensitivity of the unit to load changes is beneficially affected.

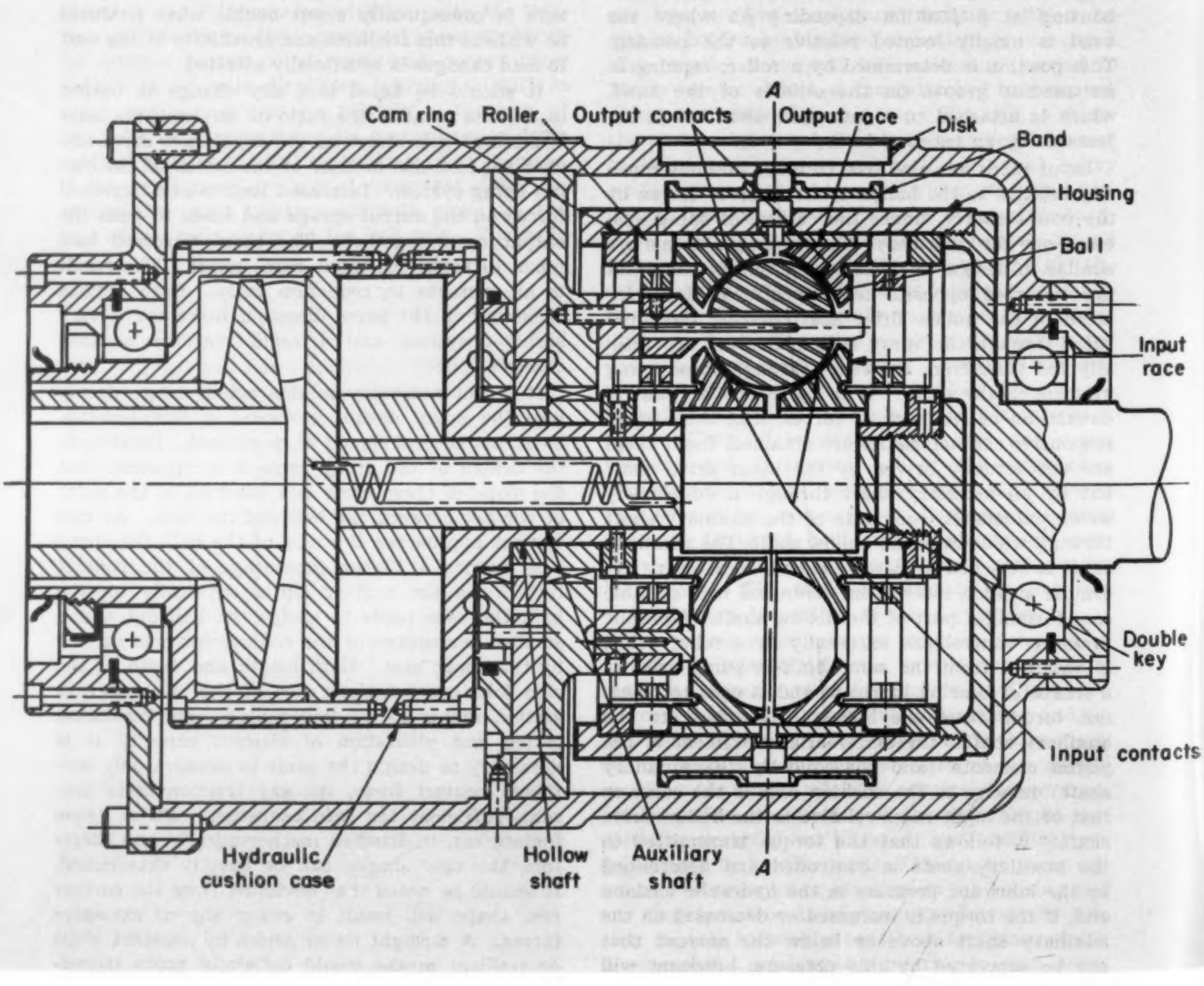
It should be noted that any change in output load tends to alter the ratio of the unit, not only because of this effect on the input race drive mechanism but, also because of the action of the output sprag system. Increased load causes increased forces on the output sprags and tends to push the output races together. The same increased load tends to move the input races apart. The result is an increase in reduction ratio. Both actions occurring in the same direction make for a very stable operation and a rapid response to load changes.

It is not necessary in this article to go deeply into the many design problems in this development, but several should be mentioned. Relative to the design of the sprag cams it is apparent that the slope of these cams is a function of the point of contact between the ball and the race. As this contact approaches the axis of the ball, the sprag cam carries a higher precentage of the contact force. As the contact shifts outwardly on the ball, the race tends to wedge the ball out and a smaller percentage of the contact force is carried by the sprag cam. Both length and depth of the cam lobe are functions also of the ratio of race section radius to the ball radius. For maximum rating and utilization of element capacity it is necessary to design the cams to produce only sufficient contact force, for any traction force present, sufficient for safe operation. All of these factors can be handled mathematically and therefore the cam shapes can be exactly determined. It should be noted that deviation from the correct cam shape will result in either slip or excessive forces. A straight linear action by constant slope or resilient means would definitely prove inoper-

ative. In the design shown, these lobes are about ¾-inch long and about 1/16-inch deep.

Whenever a chair has more than three legs there is the problem of making all legs the same length. The same problem is presented when it is desirable to have a number of contacts divide a total load approximately equally. The problem brings in the question of precision but it can be solved by correct design. Balls and races are elements that can be manufactured by present-day equipment to a high standard of precision. Balls can be matched to diameters in hundred thousandths of an inch. Races can be finish ground relative to a reference surface almost as accurately. In the case of the traction races, this is the side surface on which the sprag rollers operate. Small as these tolerances are, they are not small enough to distribute contact forces equally on all contact points because the deflections of these points, even under full loading, are so small. The loads can, however, be nearly equalized by permitting deflections in the races. This can be done by designing the race cross sections somewhat like a Belleville spring and slightly crowning the sprag rollers. Also, both input and output races are a free slip fit on shaft or housing so deflections in any direction are unrestrained except as the sprag roller contacts act as pivots. The deflections required for load equalization are not large for the element tolerances that are easily obtainable, and analysis indicates the added ring stresses are unimportant.

The question of ball load equalization brings up the general question of manufacturing precision. The ball diameters and races must be accurate, of course, but these present no greater problem than found with any antifriction bearing. Sprag rollers should be as near the same diameter as possible but again no difficulty is presented. The only other place a precision problem is present is with the sprag cams and their shoulder mountings. It is anticipated these cams can be coined and quenched in fixtures, but even if ground from solid hard rings the problem is not severe since the lobes are so shallow. In all other parts, either because of float or free fitting requirements, precision is not a problem. For example, in order to obtain a sufficient flow of lubricant from the ball axle



bearings a relatively free fit is required; consequently required concentricity tolerance of the ball holes is easily obtainable. Fits in the hydraulic cushion should be quite loose for considerable lubricant leakage because such leakage is desirable for fast response to load changes. Even with high leakage, auxiliary reliefs are built into the cushion for fast evacuation on sudden overload, rather than relying on the externally adjustable control.

Rating of a unit, as described, is a function of several factors. As the ratio range is increased, (and built-in stops can control the ratio limits) the power capacity is decreased inasmuch as contact forces increase. Changing the ratio range from 8-to-1 down to 6-to-1 increases power capacity about one third. Ball diameters have considerable effect. For a given number of balls, not only do curvature diameters increase with the ball diameter, but traction moment arms increase throughout the unit. It is probable that ratings are roughly a square function of overall size. Localized wear due to prolonged operation on a given setting controls practical ratings for industrial applications on most such devices. For applications where load conditions cause almost continuous ratio change, ratings can be definitely increased in this torque-responsive design.

The design is presented as being particularly adapted for industrial type drives for ratio ranges of about 6-to-1 and horsepower ratings up to about 30 hp. The unit will always start at maximum reduction ratio regardless of speed adjustment, because in the absence of lubricant pressure the ratio changing elements move to this position. High-inertia loads therefore can be started without strain or special drive motor torques. As the inertia is overcome, the output shaft will come up smoothly to the speed setting. Torque load on the drive motor can be set for any value and will remain constant throughout the ratio range.

A very interesting application possibility for this development is its use as an automatic transmission in automotive drives. The automotive engine produces a torque that is high at low speeds,

Fig. 15—Left—Cross section through axis of torque-responsive transmission

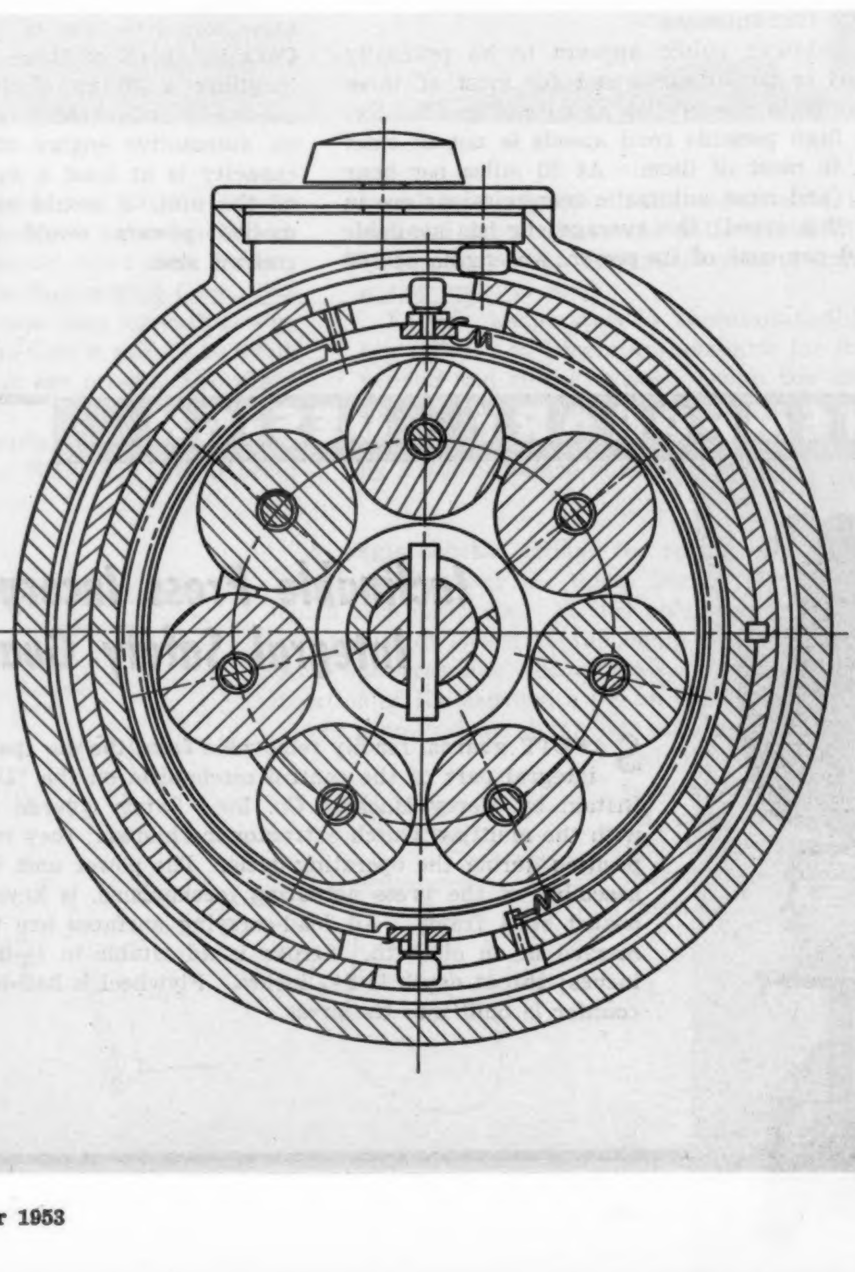


Fig. 16 — Right — Cross section through centerline *A-A* of *Fig.* 15

gradually lower with increase of speed until at high speeds it falls quite rapidly. The output horsepower increases with speed to a maximum and then falls off. At some one speed it produces maximum power and the torque at this speed is below the maximum torque value. Consider the effect on available power to the rear wheels if a transmission would load an engine to maximum power torque throughout all ratios. The difference between this torque and that obtainable at speeds under maximum power speed is available for acceleration of the engine and full engine power is therefore at the call of the car regardless of road speed. With 4-to-1 reduction, plenty of torque at the rear wheels is obtainable and this torque can be obtained throughout practically the entire city driving speed range. When going down a hill or cruising, a 1½-to-1 speedup allows the engine, still pulling at full torque, to run more slowly. Top car speed would be controlled by peak power. Transmission efficiency of 90 per cent can be expected, possibly higher, through all ratios. Not as high as direct drive, it still is a better average than present automatic transmissions, except when they lock into practically a direct drive at higher speeds. However, at that time they are no longer automatic transmissions.

The American public appears to be primarily interested in performance and for most of them performance is acceleration at normal speeds. Extremely high possible road speeds is not of chief interest to most of them. At 30 miles per hour in high (and most automatic transmissions are in high at this speed) the average car has available about 40 per cent of its power. An engine of 100 hp at that reduced speed produces possibly 40 hp. If full power were available at this speed by the use of a torque-responsive drive, a 60-hp engine would outperform the 100-hp engine.

Gasoline consumption is greatly affected by percentage of load. At idling speed most engines consume about a quarter as much gas as at full load. Efficiency improves rapidly with load but is very low at a quarter of full load, still low at half loading and best efficiency is obtained at three quarters of loading or more. Consider the effect on gas mileage if the engine torque could be maintained at full value a large percentage of the time.

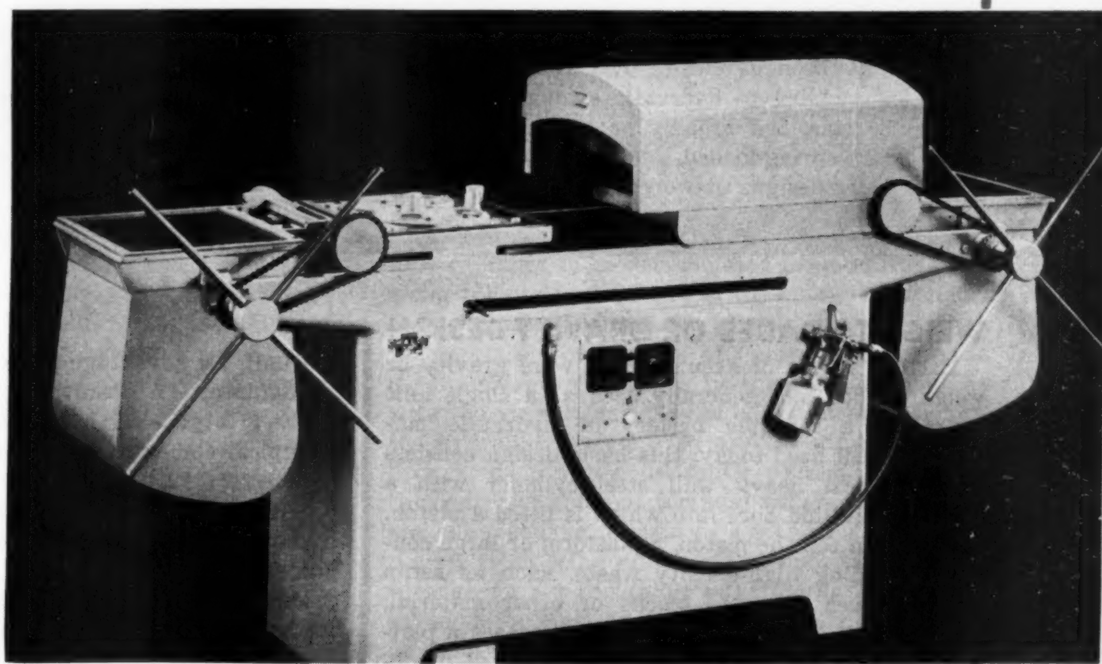
The traction type unit could be rated on a different basis for automotive service than for industrial drive service. Constantly changing ratios would distribute wear, for one thing, and maximum torque available from the engine, being only a little more than normal setting, would allow a higher rating than if the unit were driven by an electric motor with possibly 200 per cent torque occasionally in use. Low gears in automotive transmissions are designed for a very short life at full load because they seldom handle full load and are used such a low percentage of the time. The same would be true of the traction transmission. Considering all of these factors a unit capable of handling a 30 hp electric motor for industrial use could undoubtedly handle, with adequate life, an automotive engine of 60 hp or more. Since capacity is at least a square function of the size of the unit, it would appear that present automotive powers would not require greatly increased size.

CONTEMPORARY DESIGN

Inclinable Press Incorporates Integral Safety Guards

SAFETY guards, readily removable from the die space of the machine, are an integral part of the control mechanism on the "Unit" press imported from Britain by Morey Machine Co. Inc. Safety guards are positively interlocked with the multijaw clutch extractor mechanism; they remain locked in the closed position during the operating cycle. The power unit, a complete, self-contained assembly of the press actuating mechanisms, is keyed to the welded and annealed steel frame. All load-carrying surfaces are totally enclosed and submerged in an oil bath. Stroke is adjustable in ½-inch steps from ½ to 2½ inches; throat depth is 5¼ inches. Flywheel is ball-bearing mounted. A work counter is built into the press.

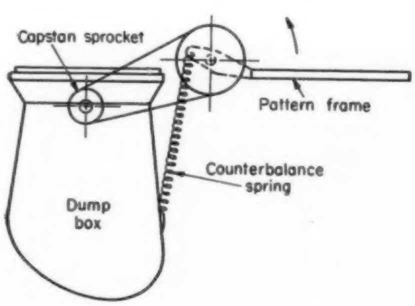
Shell Mold Machine Features Simple Operation



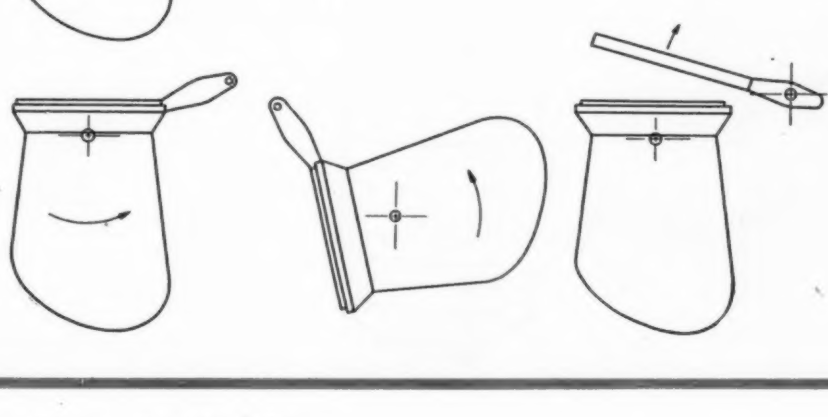
DUAL dump boxes and pattern frames permit two patterns to be handled alternately on the model HO-4 shell molding machine of Shallco Engineering Corp. Simplicity of design and easy operation were two objectives; the unit is said to be available at low cost yet can produce two shells per minute.

Two different curing ovens, gas or electric, are available. While one shell is being cured on its pattern, the cured mold is stripped from the second pattern and the pattern and dump box are inverted to form a new uncured mold.

Pattern temperature is thermostatically controlled, and timers are available for inversion and curing cycles. Dump box design is said to provide even sand fall and dense, uniform shells. Sheils are ejected by a self-contained pneumatic stripper.



A single capstan swings the spring counterbalanced pattern over the dump box by means of a chain and sprockets. At this point the pattern is automatically locked to the dump box and sealed with heat-resistant silicone seals. With continued rotation of the capstan, a pin on the capstan shaft meets a dog on the dump box and inverts the assembly. The pattern is then swung back to its original position and the curing oven moved over the uncured mold.



HYDRAULIC ACCUMULATORS

VARIOUS types of accumulators have been developed over the years and applied to hydraulic systems. These can be classified broadly as: (1) Weight-loaded, (2) spring-loaded, and (3) pneumatic. The older designs of weight and spring-loaded accumulators as well as those of the simple air-bottle type can be further subdivided as non-separator type as compared with the later designs employing a special diaphragm to separate the hydraulic fluid from the pneumatic loading gas.

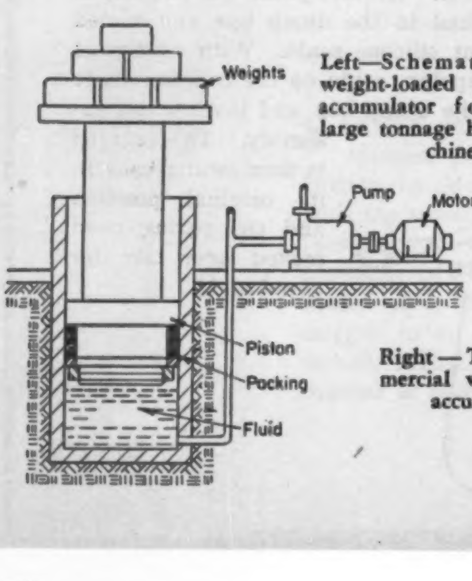
WEIGHT-LOADED OR GRAVITY DESIGN

Earliest forms of accumulators were gravity or weight-loaded and generally used as a single unit operating a multiple battery of hydraulic machines. Still used today, this basic design consists of a vertical heavy-wall steel cylinder with a smoothly ground bore into which is fitted a piston. Mounted on top the piston, a platform or large container holding high-density waste, such as scrap metal, stones, concrete blocks or other material, provides the counterweight. In some cases a platform container filled with water is utilized to obtain the required weight. With these accumulators force of gravity provides the energy. Constant pressure is available from the full-volume output of the unit, an advantage possessed by no other type accumulator. A disadvantage, however, is the extremely large size, heavy weight, and high cost. A typical commercial weight-loaded unit having a capacity of 1200 cubic inches weighs 3500 pounds plus 35,000 pounds of weights. An equal capacity hydropneumatic 10-gallon accumulator weighs 170 pounds.

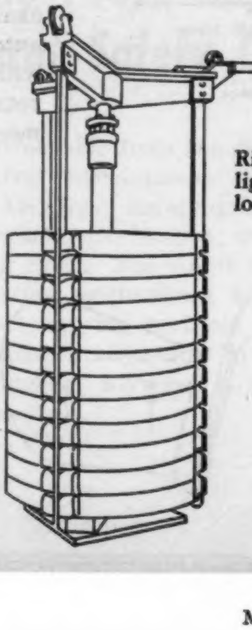
Due to the large size and high cost of the weight-loaded type, it is essential that a single accumulator be used in a plant operating multiple machines. Should the accumulator fail for any reason, all the hydraulic equipment in the plant would be incapacitated.

SPRING-LOADED DESIGN

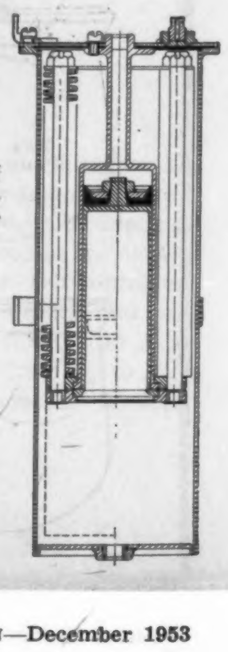
A spring-loaded accumulator consists essentially of a cylinder into which is fitted a piston arranged so that compression springs supply the dynamic load. Located about the periphery of a disk mounted on the end of the piston is a series of axially-fixed rods supporting a similar number of compression springs. The rods pass freely through a second disk attached to the cylinder against which one end of the springs bear. The opposite ends of the rods retain the other end with adjustable nuts. The springs are compressed to correspond with the minimum pressure required in the circuit by tightening the adjusting nuts. As fluid is pumped into the chamber, the piston rises, further compressing the springs and loading them to correspond with the increased pressure developed.



Left—Schematic view of weight-loaded gravity type accumulator for operating large tonnage hydraulic machinery



Right — Typical commercial weight-loaded accumulator



Right — Section of light-duty spring-loaded accumulator

2—Nonseparator Designs

By Edward M. Greer

President and Chief Engineer
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The volume of accumulated fluid at pressures corresponding to any position of the piston is dependent upon the rate and loading of the springs.

One commercial version of the spring-loaded accumulator, 8 inches in diameter, 20 inches long, weighing about 75 pounds, has a fluid capacity of 40 cubic inches and supplies oil at pressures between 750 and 1500 psi. A typical hydropneumatic type is 4 inches in diameter, 6 inches long, and weighs 16 pounds.

PNEUMATIC DESIGN

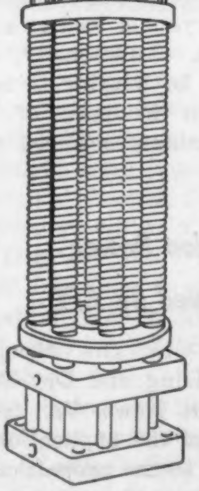
Pneumatic accumulators have proved to be much more practical than the weight and spring-loaded types. They may be classified into nonseparator and separator types. The former has no means of sealing the gas from the fluid. As used in industry today, it generally takes the form of an air bottle. Gas, either air or nitrogen, is trapped under pressure on top of the liquid. In the simplest form, it is known as an air bell, located on top of the discharge port of duplex and triplex pumps. It is merely a hollow shell in which atmospheric air has been trapped at the highest point of the pump. In time, air is absorbed into the fluid making regular recharging necessary.

A high-pressure nonseparator type accumulator generally has a fully-enclosed shell containing a fluid inlet port at the bottom and a pneumatic charging valve on top. To place such a nonseparator type accumulator in service, it is necessary to fill the unit partially with hydraulic fluid and charge compressed air or nitrogen through the pneumatic valve to the minimum pressure requirements. As additional hydraulic fluid is pumped into the chamber, the gas is compressed further in accordance with Boyle's law—The volume of a given mass of gas at constant temperature varies inversely as the pressure. In this way a large volume of fluid can be accumulated to do useful work using the compressed gas as the energy medium.

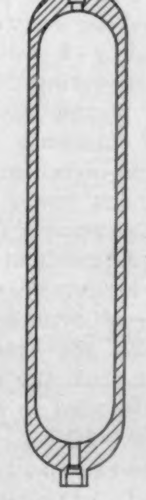
One great disadvantage of this type of unit is that aeration, or mixture of fluid and gas, results in diminishing precharge in the same manner as with the air bell, and results in erratic operation of the hydraulic system. To prevent air from being exhausted into the hydraulic system, only two-thirds of the accumulator fluid volume can be used and the rest must remain as a separator to prevent the air from being drawn out at high rates of flow through an air vortex. Furthermore, this type of accumulator must be mounted in a vertical position as the gas must be trapped at the top of the cylinder. In installations where hydrocarbon oil must be pumped under high pressure, the diesel effect, which can be described as the rapid increase of pressure resulting in explosion, can be disastrous.

Multiple installation of large nonseparator type accumulators is common practice where extremely large volumes of 500 gallons or more are required.

Equipment required in addition to the usual air bottle accumulator is high-pressure air compressor, pressure-regulating valves, and controls. The compressor operates against a level regulator to ensure that the discharge valve is completely cut off at a safe low level.

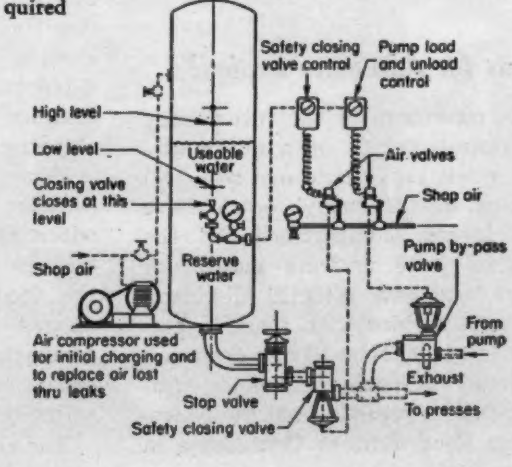


Left — Heavy-duty spring-loaded accumulator of 40 cubic inches capacity



Right — Gas bottle nonseparator type accumulator

Nonseparator type accumulator circuit showing accessories required

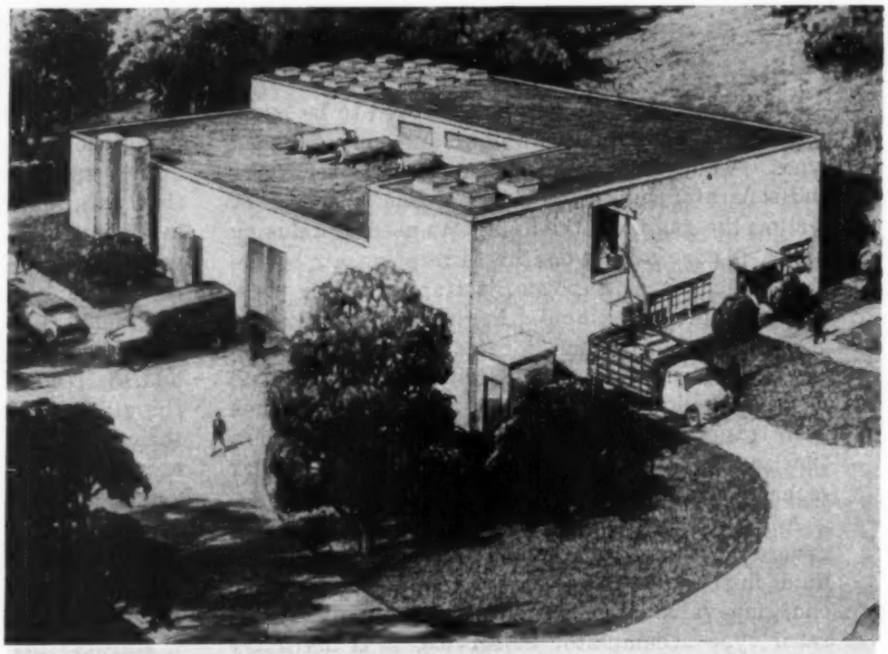


Engineering NEWS ROUNDUP

Columbium-Tantalum Restrictions Removed

Last remaining restrictions on the use of columbium and columbium-tantalum for the production of civilian goods were removed by the Business and Defense Services Administration, Department of Commerce, with the revocation of Order M-106, effective November 1, 1953. In announcing revocation of the order, officials of BDSA said control is no longer necessary because supply substantially exceeds the demand, and military requirements have been reduced. They also stated that adequate safeguards have been made to assure availability of required amounts for the national stockpile.

Columbium and columbium-tantalum, the major U. S. supply of which is imported, were placed under government control on October 19, 1950 with isssuance of NPA Order M-3, which prohibited the use of ferrocolumbium bearing steels in any application or process where other steels could be substituted, or use of the alloys where substitutes could be used. These alloys are used principally as a stabilizing agent for certain stainless steels which require welding in their application.



COMBUSTION LAB: Scheduled for completion late in 1954, this one and one-half story building, with 14,200 square feet of floor space, will be devoted to basic scientific studies to determine the exact chemical and thermal nature of combustion. Knowledge of combustion processes gained in the $1.8 million General Electric lab will permit production of more powerful gas turbines for aircraft and other applications

Ideas for Appliance Design?

Refrigerators of the future may be atom-powered, open automatically when an electric eye sees you coming, heat the baby's bottle, have the freezer compartment at the bottom, have drawers instead of doors, and have integral all-plastic cabinets molded with foam insulation in place, Mrs. Vivian Overand, director of home economics, Admiral Corp. recently told the Newspaper Food Editors Conference in Chicago.

Mrs. Overand pointed out that 100 to 125 million pounds of plastics will be used in the construction of approximately 4 million electric refrigerators during 1954. Advantages of the plastic are resistance to acids, chemicals and chipping, light weight, excellent insulating properties, low water absorption, and nonabsorption of food odors. Freezer compartments at the bottom are quite logical because the food storage compartment is opened oftener than the freezer compartment. Heat from the condenser might well be used to heat water or the baby's bottle.

The electric range has also stimulated Mrs. Overand's imagination. For future ranges, she foresees combination built-in pressure cookers and deep fat fryers as well as automatic toasters. Additionally, she feels it might be feasible to mold the circuits in the back of the range and eliminate soldered connections.

Ship Propulsion Power Study Approved by AEC

A proposal by the Newport News Shipbuilding and Dry Dock Co. of Newport News, Va., for a special study of the application of nuclear power to the propulsion of ships has been approved by the

Atomic Energy Commission. The company developed experience in the nuclear field as a subcontractor on the now terminated project for an aircraft carrier reactor. Six of its engineers recently completed studies at the Oak Ridge, Tenn., School of Reactor Terminology.

All costs will be borne by the company. The contract will run for one year after which a complete report of findings and recommendations will be submitted to the Commission. After full investigation, a limited additional number of NNSDD personnel will be cleared if necessary to investigate pertinent phases of the Commission's nuclear power program.

Lightweight Gas Turbines

Two new lightweight gas turbines feature complete enclosure of hot parts of the turbine and nozzle box in a pressurized chamber. This feature is said to eliminate a design problem found in earlier turbines. In the event that the outer skin is pierced or cracks due to vibration, leakage consists of compressed air at the heat of compression rather than hot gases at combustion temperatures.

One of the turbines is a compressed air supply unit weighing 170 pounds and having an output rating roughly equivalent to 170 horsepower. The other is a combination compressor and shaft power unit in the 120-horsepower class. The units are manufactured by AiResearch Mfg. Co.

Engineers, Educators Propose Changed Engineering Education

Higher standards for accreditation of engineering programs in colleges and universities were recommended by the American Society for Engineering Education and the Engineers' Council for Professional Development at a joint meeting on October 16. The proposal called for division of engineering education into "professional-scientific" and "professional-general" education.

The recommendation came from a committee appointed last year because "only minor adjustments have occurred in engineering education" to match advances in the physical sciences in the last 10 years. These advances, coupled with anticipated availability of nuclear power, "necessitate major changes in the character of engineering education," declared L. E. Grinter, dean of the Graduate School, University of Florida and president of ASEE.

"Professional-general" programs, Dean Grinter said, would train engineers to serve in areas between engineering and business, management, law, real estate, or agriculture; between engineering and a branch of science with limited application of engineering analysis and design; and between engineering and such highly applied technologies as air conditioning or welding. There is a growing need for men in general professional engineering sciences, Dean Grinter said.

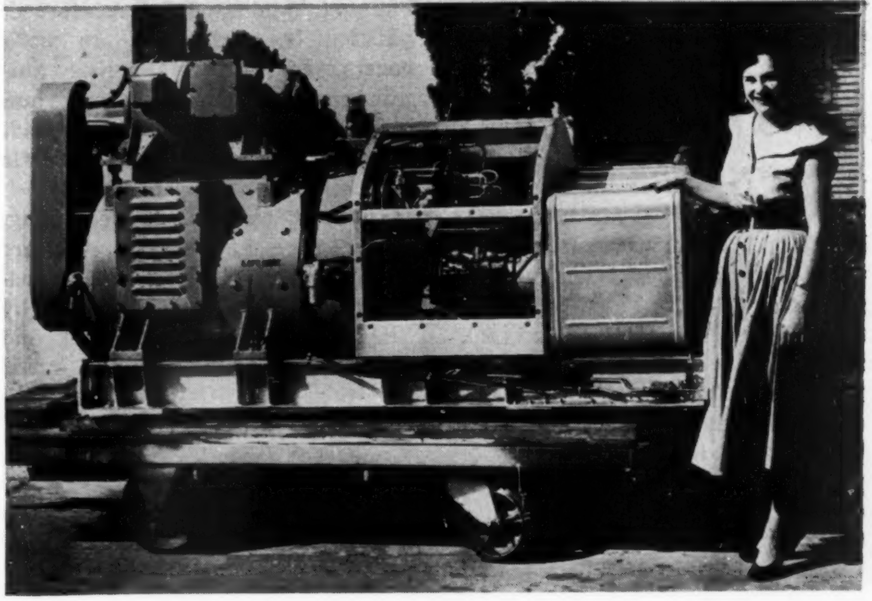
Four-year programs may not be sufficient to meet the committee's recommendations for the professional-scientific curriculum plus the specialized courses of the degree-granting department, the speaker declared. But the committee believes, he said, "that specialized engineering courses are of far less value in professional-scientific education than a broad background of engineering science."

Raised professional - scientific standards, Dean Grinter said, would include: increased background in mathematics, physics, and chemistry; study of nine engineering sciences; and extension through four semesters of the study of engineering analysis and design, or of engineering systems.

The committee also proposed special designation for institutions with standards substantially above the minimum. Faculty eminence—the presence of more than the minimum number of distinguished members—would be the major factor in conferring special designation, Dean Grinter said. The committee's minimum requirement for faculty is one member in five "who has achieved professional distinction by creative activities."

Social studies and the humanities should be centered outside such courses of immediate usefulness as accounting, industrial psychology, and city managements, the committee concluded. Instead, humanistic-social studies should afford "a clearer perspective of civilization, inspiration for seeking

SEAGOING GAS TURBINE: This shipboard electric generator set, is powered by a 500-horsepower gas turbine said to be the most powerful installed in an American Naval vessel. Weight of the Solar Aircraft Co. turbine, with accessories, is about 750 pounds; length is about 6 feet; and height and width are less than 3 feet



SUPERSONIC SABRE: First picture of the F-100 Super Sabre shows 45 degree sweepback of wing and tail. First production model of this 45-foot long 36-foot span North American fighte: came off the assembly line on October 20. Combat radius is over 500 nautical Miles, service ceiling is over 50,000 feet, and speed is supersonic

greater knowledge and understanding, and aid in development of judgment and discrimination, a sense of value, and sound personal philosophy."

The fact that in the last 10 years nuclear problems have occupied an increasing number of research physicists, Dean Grinter pointed out, is a major factor in the need for revising engineering education. "It seems doubtful," he said, "if the interests of these physicists will ever be returned in sufficient measure to influence greatly research in vibration, elasticity, heat transfer, and the other engineering sciences." Engineers, therefore, have become responsible for research in these fields.

Machine Tool Builders Ask Fewer Restrictions

The Machine Tool Manufacturers Industry Advisory Committee recently recommended amendment of government order M-41, to permit manufacture of more machine tools for nondefense uses. The order provides that 60 per cent of machine tool output be used to fill defense rated orders, leaving 40 per cent for nonrated orders. The committee recommended reversal of these percentages, in view of slackening defense orders.

Also recommended were: elimination of a critical list of machine tools on which the split is 70 per cent for defense and 30 per cent for nondefense; elimination of the requirement that order boards be submitted for inspection before proceeding with production, and increasing the present three month freeze on production schedules for certain machine tools requiring long lead time to six months.

Plastics Used In Trailer Manufacture

Large trailer bodies are being built of laminated or reinforced, plastics by companies on both coasts. Lighter weight than aluminum, corrosion resistance and dent resistance are among the advantages claimed for the plastic bodies.

Complete bodies 24 feet long, 8 feet high and 8 feet wide are being built by Seaboard Transportation Co. of California by "laying-up" glass fiber cloth and mat in wood, steel or plastic molds and then saturating the mat with Vibrin polyester resin, made by Naugatuck Chemical Div. of U. S. Rubber Co. Performed ribs of the same material are pressed into the wet lay-up to form a complete section which becomes an integral unit when cured.

Transparent plastic skylight panels, molded plastic doors, door frames and interior liners are used in trailer construction by the Strick Co. of Pennsylvania. Complete sides and roofs of a number of van-type trailers are also made by this company. Glass fiber and Vibrin are used to make these parts by a vacuum impregnating method in which the plastic resin is sucked into a mold in which the glass fiber has previously been placed.

Awards for Welded Designs Made to Machine Designers

Engineers and designers in all parts of the country recently received 77 awards totalling $30,000 for welded machine designs entered in a Mechanical Design Award Program sponsored by the James F. Lincoln Arc Welding Foundation. Dimitri Soussloff, associate director of research, Universal Winding Co., received the first award of $2981 for his description of a yarn twisting machine, parts of which were redesigned for welded construction. It is estimated that the new design will save $100,000 a year.

Awards were made for designs covering a wide variety of machinery, including metal cutting, electrical, conveying and pumping, compressing, tooling, processing equipment, and others. Second award of $2882 was to David W. Kinney, chief engineer, Pattin Mfg. Co., for a special machine to eliminate an unsafe and expensive hand grinding operation on wedges used in bolting mine roofs. Robert J. Neville, vice president in charge of manufacturing, North American Mfg. Co., received the $2385 third award for the design of welded industrial type gas and oil burners.

Ralph E. Flanders received the 1953 Howard Coonley Gold Medal for service in advancing the national economy through voluntary standards at the annual award luncheon meeting of the American Standards Association at the Wal-

dorf-Astoria, New York, October 21. Senator Flanders is a past president of ASME and the National Machine Tool Builders Association; he is also a former vice president of the American Engineering Council. Until becoming senator in 1946, he was president of the Jones and Lamson Machine Co., Springfield, Vt.

New Flight Structures Lab

Establishment of a new educational and research unit for the study of air flight structures, particularly in the supersonic range, was announced recently by Columbia University and the Daniel and Florence Guggenheim Foundation, which has provided a grant of $329,000 to establish the new center. To be known as the Daniel and Florence Guggenheim Institute of Air Flight Structures, it will be organized within the Columbia University Engineering Center. The new lab will begin operation on January 1, 1954.

Purpose of the lab will be fourfold: to train exceptionally qualified graduate students in the comparatively new field of air flight structures; to conduct research in aircraft structure and design, especially for supersonic flight; to act as a national clearing house for technical information in this field; and to disseminate technical knowledge regarding air flight structures.

An advisory board, consisting of nationally known figures in aviation and engineering, will be appointed to assist in guiding the Institute's activities.

Tape Recorder To Operate Milling Machine

A record-playback control which directs a machine or process cycle from information stored on a magnetic tape is being built to control a milling machine that manufactures self-reinforced "skin" structures for jet aircraft. The industrial type multiple-channel tape recorder used in the control is similar to equipment employed in the entertainment and broadcasting fields, except that it records motion instead of sound. It will serve as a magnetic memory for the machine on which it is installed. Builder of the control, General Electric Co., expects to deliver it to Giddings and Lewis Machine Tool Co. in time to permit the completed skin milling machine to be demonstrated to aircraft manufacturers early next year.

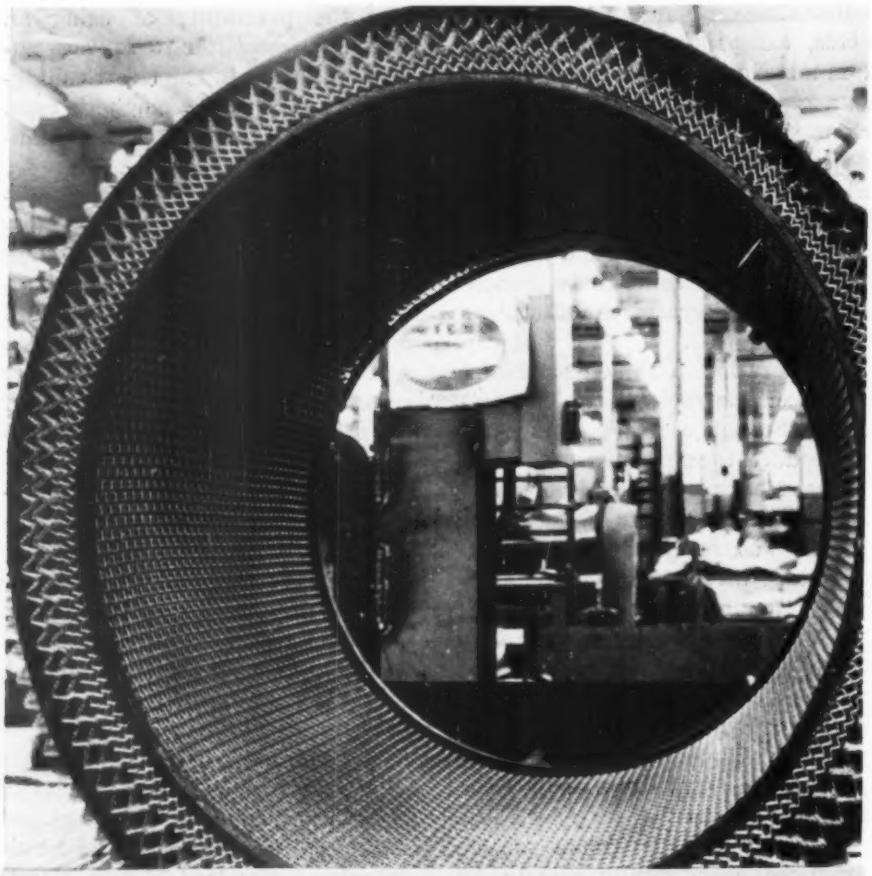
In operation, a skilled machinist or tracer control will run the machine through a series of motions to produce the first piece of work. These motions will be translated into electrical signals by means of selsyns and recorded on the magnetic tape through an electronic circuit. Nonproductive motions such as tool setting, gaging, template changing and stop settings are not recorded. When the tape is played back through the machine it automatically duplicates the machine's original motions and produces a part identical to the one made while recording. Repeat orders can be produced economically merely by taking the specified tape recording from a file and inserting it into the control.

Research studies are now being conducted by Giddings and Lewis into the possibility of using an electronic computer to convert numerical data into the electrical signals recorded on the magnetic tape. This would eliminate the first manual machining operation.

While the record-playback control may be used in many applications other than the skin milling machine, General Electric engineers point out that the control must be individually engineered to suit each installation and carefully co-ordinated with the design of the machine it is to control. In addition, the equipment is relatively expensive, so many possible applications are eliminated at the present time by economic considerations. However, any motion or control that can be operated from an electric signal can be operated from the record-playback control, they explained.

The length of cycle that can be controlled depends on the diameter of the reel of tape and the speed at which it is recorded and played back. On the Giddings and Lewis machine a standard 14-inch diameter reel is expected to run a one-hour cycle. When desired, recordings can be erased easily, restoring the tape to its original condition for a new recording. The tape may

IDEA CAR: Shown at the Paris International Automobile Show recently was this Chrysler Special sports coupe. Built on a Chrysler New Yorker chassis and powered by a Firepower V-8 engine, the body was designed by Chrysler and built by Ghia of Turin, Italy



GIANT FILTER ELEMENT for a huge oil-bath air cleaner measures 36½ inches OD, 27½ inches ID and is 30 inches high. Filters such as this with its 6500 cubic foot per minute capacity, manufactured by the Air-Maze Corp., are for service such as cleaning intake air for diesel engines and compressors

be cut and new sections spliced in if only part of the work cycle has to be changed. There is no deterioration of the signal when the tape is stored and play-backs up to 10,000 times are possible without appreciable loss of controlling signal.

Record-playback equipment possesses certain inherent advantages over existing methods of automatic machine and process control. It offers complete co-ordination of all machine or process variables such as continuous control of feeds to produce desired contours, multiple passes if required, change of spindle speed to suit desired machining conditions at any time, and synchronization of many other functions of the cycle. Additionally, operator know-how can be incorporated into the recorded automatic cycle for such things as compensating for backlash, taking care of machine inaccuracies, spindle deflections, tool loading, and so on.

The control is said to make possible high production economies for low or medium production jobs. Automatic and semiautomatic cycles can be used where they formerly were impractical. Minimum set-up time is required when changing from one automatic cycle to a totally different cycle since only the tape and tools need changing.

The equipment itself consists of two components: the magnetic tape recorder and the electronic control panel. Both are enclosed in a cabinet about the size of a large wardrobe closet. Weight is approximately 1900 lbs and power requirement is standard.

Engineer Says Check Economics of Automation

Top management must attack the problems of the automatic factory with "grammar school arithmetic and horse sense," brushing aside the higher mathematician as well as the science fictioneer, the sixth Industrial Management Institute of the University of Illinois was told recently.

The advice came from George M. Muschamp, vice president in charge of engineering, Brown Instruments Div., Minneapolis-Honeywell Regulator Co., whose topic was System Engineering. This he described as a way of cutting through the maze of means offered to maintain more automatic production, by keeping one's eye on the end problem of more and better goods at less cost.

"In the further stages of development there will be engineering, scientific, financial and other business considerations of great complexity," said Mr. Muschamp, but he urged that these be subordinated to management's prime purpose of operating at a profit.

"Marvelous computers, clever conveyor belts and multipurpose automatic machines," the session was told, must not be allowed to obscure economies, maintenance problems, or output capacity versus sales requirements. Automatic machinery and its techniques, Mr. Muschamp warned, must be assimilated by the staffs operating present factories. It is too much to expect, he said, that the technicians, who have done far more than their share in developing these technological tools, must also learn the intricacies of each individual industry.

Frank G. Steinebach, editor, *Foundry* magazine (a Penton Publication) was honored at a special luncheon session of the Gray Iron Founder's Society during the Society's annual meeting, held recently at St. Louis. Mr. Steinebach received the Society's highest award—a gold medal—for his outstanding personal and professional contributions to the general wel-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fare of the industry and his services in connection with the formation and maintenance of the National Castings Council.

Recommendations Made For Engineer Utilization

During the National Manpower Conference on the Utilization of Scientific and Professional Manpower, held recently at Columbia University, the engineering group, consisting of 18 engineers, engineering educators and engineering personnel, agreed on a number of procedures to improve engineer utilization. Improved utilization was defined by the group as any step which increases the quantity or quality of goods and services produced with a given amount of capital, using a fixed number of technological personnel, or, alternatively, as any step which maintains a given level of output in terms of quantity and quality, with a smaller number of such personnel.

The following points were agreed upon as steps to follow to secure improved utilization:

1. Make more effective utilization the focus of genuine and continuing interest at all levels of an enterprise.
2. Efforts to improve utilization must distinguish among what can be done to produce immediate gains, what may yield results only after some investment of time and money and what changes are fundamental and long range in character.
3. If there is any single point at which improved utilization is initiated, that point is in training, not only in colleges and on the job, but also in elementary and secondary schools.
4. Jobs which engineers are doing should be studied to learn what duties might be reassigned to other technical personnel.
5. Engineers must be provided with the best tools, instruments and supporting personnel, if they are to make efficient use of their training and talents.
6. Effective communication between management and engineering personnel about the objectives of their work is necessary to improve utilization.
7. Organizational changes may permit improved utilization because too much organization may prevent necessary freedom of effort, while too little organization may result in duplication, lack of proper priority, and unwise allocation of personnel to projects.
8. Incentives in the form of status, recognition and money rewards generally associated with jobs in management and sales can improve utilization by keeping technologically trained people at the technological jobs for which they are best suited.
9. Recognition of the fact that many things outside an individual's job affect his effectiveness can improve utilization.
10. Due regard must be had for a normal human resistance to change where an improvement in utilization is attempted.
11. Management can improve utilization by calling upon the manifest capacities of engineering personnel and by making an effort to develop potential capacities by imaginative assignment and training.
12. No single device will improve utilization; many related measures must be employed.

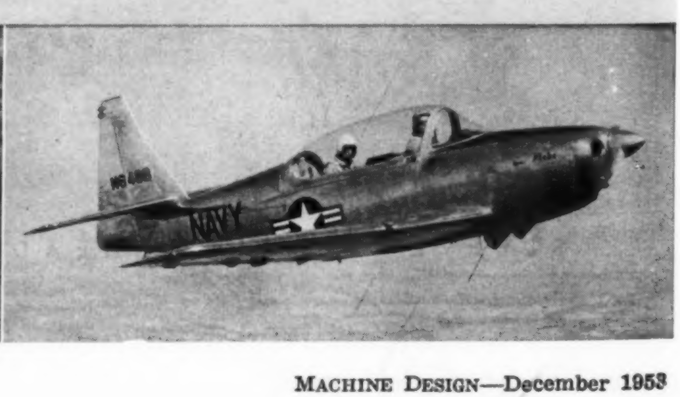
Continuous Steel Casting May Be Around the Corner

A recent announcement disclosed that a machine for continuous casting of steel is being developed. Continuous casting, the casting of billets, slabs or rounds directly from the molten metal, completely bypasses pouring ingots, heating in soaking pits, and the first steel-mill rolling operation—rolling on the blooming mill.

Four years of pilot plant operation convinced officials of the Allegheny-Ludlum Steel Corp. of the feasibility of the process. Although there is no commercial continuous casting plant for steel in operation today, the process has long been used in the nonferrous metals industry. The pilot plant installation was patterned after a similar unit used in the brass industry.

Equipment being considered by Allegheny-Ludlum has 150 square inches of mold cross-section capacity. It is hoped that a casting

COMPETITORS: Entries in a design competition and evaluation test to choose a trainer for fledgling Naval pilots are the Temco Plebe, right, and the Ryan Model 72, left. Developed in 75 days to enter the competition, the Plebe has a service ceiling of 20,000 feet; 1350 fpm rate of climb; top speed of 185 mph, maximum endurance of 5¼ hours, and weighs 2500 pounds. The Model 72 has a side-by-side seating arrangement for student and instructor, and is designed to use either a Continental or Lycoming engine. Wing span is 37 feet, 8 inches



(Advertisement)

Facts about *HELI-COIL* inserts you should know

What they are

*Heli-Coil** screw thread inserts are precision formed coils of stainless steel or phosphor bronze wire. Wound into tapped holes, they form permanent, non-corrosive, strip-proof threads of astonishing strength. Available for National Coarse, National Fine and Unified threads, pipe threads and spark plug threads. They are made in all standard sizes and lengths for assemblies requiring Class 3, 3B, 2 or 2B fits.

What they are for

AS ORIGINAL COMPONENTS: *Heli-Coil* inserts are used to provide stronger, lighter fastenings, corrosion-proof, wear-proof threads in all assemblies.

FOR PRODUCTION SALVAGE: When conventional tapped holes are damaged in production, restore them on the line with *Heli-Coil* inserts. Get better-than-original strength with no increase in screw size and no tell-tale signs of rework.

FOR SPEEDY REPAIRS: When tapped threads wear, strip or corrode in service, renew them in minutes on location in shop or field with *Heli-Coil* inserts. No welding—no plugging—no secondary machining—no oversize screws.

How they work

Holes are drilled and tapped as you do for ordinary threads—then *Heli-Coil* inserts are wound into tapped holes by hand or power tools. Install in a few seconds, assure thread protection forever. Can be used in any metal wood or plastic.

No other method is so simple, effective and practical.

What they do for you

Heli-Coil inserts *save money* because they strengthen threads and make *fewer* smaller fastenings do the same holding job. They make *lighter* bosses and flanges practical and they *save weight* in two ways: (1) by permitting use of cap screws, instead of bolts and nuts; (2) by allowing use of smaller, shorter, fewer cap screws. *Heli-Coil* inserts *protect* your product from thread wear, galling and stripping for life in every kind of metal, in plastics or wood. They preserve customer good-will by preventing product failure, due to thread fault. *Heli-Coil* inserts improve the end product, cut rejects, salvage threading errors.

Best time to put *Heli-Coil* inserts benefits to your use is right at the designing board, as many leading manufacturers are doing. But to convince you of their many advantages ask for a working demonstration right on your production line. Write today! Complete information and engineering data is available in the *Heli-Coil* catalog. Use Coupon!

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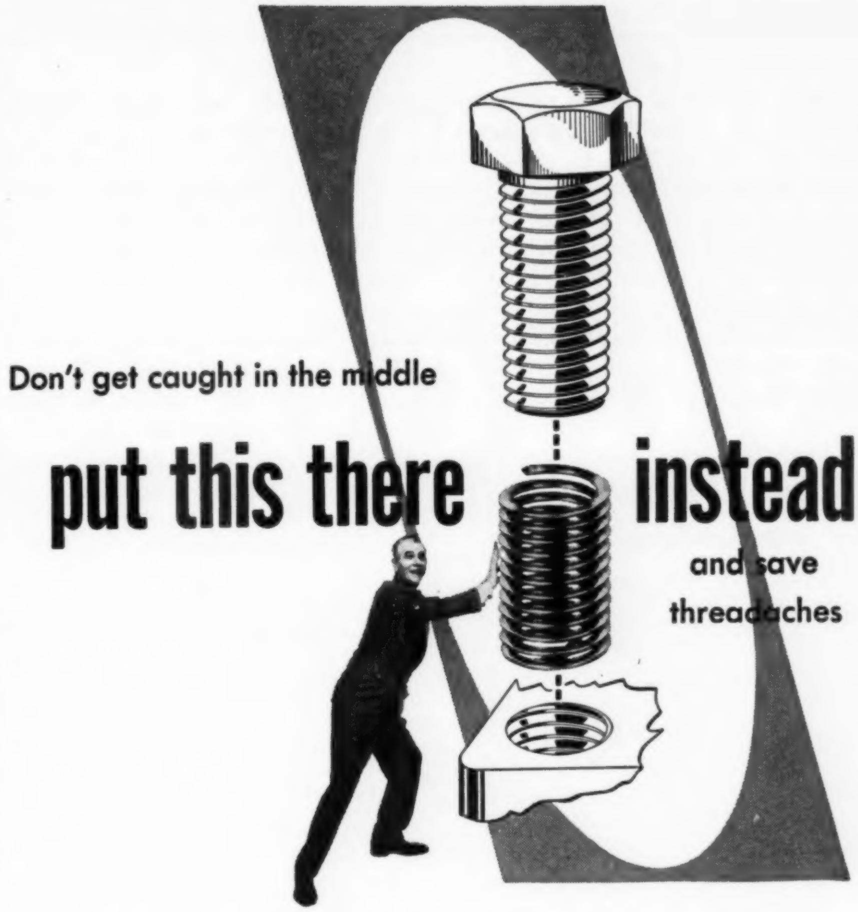
Heli-Coil Inserts conform to official military standards MS-122076 (ASG) through MS-124850 (ASG) and others.



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speed of about 120 inches per minute will be possible. Specifications call for the unit to produce up to 20 tons per hour as a single-strand caster, and double that amount when operating as a two section unit. Continuous Metalcast Corp., holder of basic patents in the field, is assisting in the design work, as is the Freyn Engineering Div. of the Koppers Co.

Certified Teflon Reports Are Now Available

Users of Teflon can now be assured of specified performance with "electrical grade" rods, tubes, sheets and finished parts by a system of quality control and certification instituted by Resistoflex Corp.

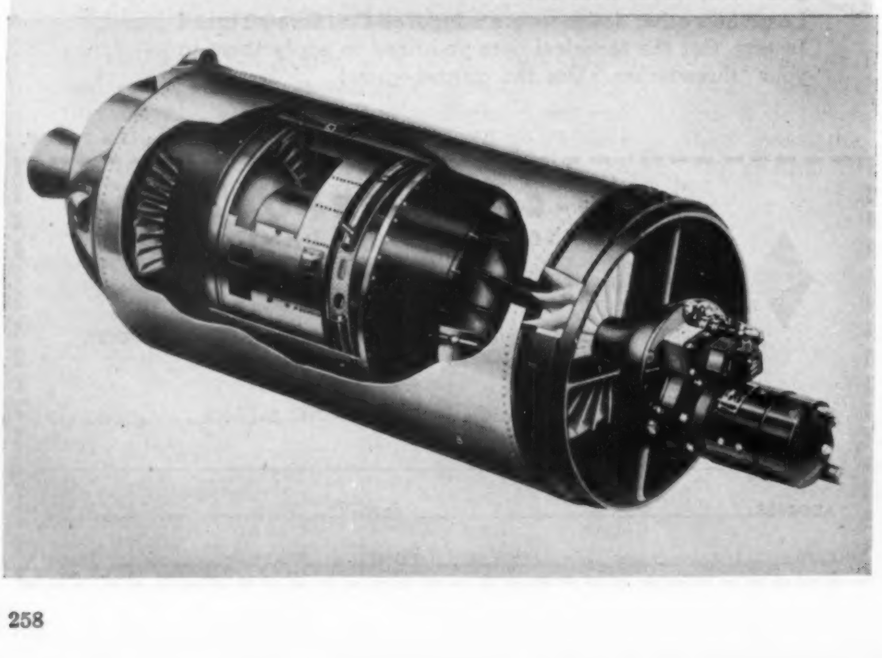
For the first time a user can obtain test reports on a particular lot of Teflon. Certification is based on six qualification tests carried out by the company in conjunction with U. S. Testing Laboratories. The tests cover power factor, dielectric constant, dielectric strength, porosity, tensile strength and elongation. These tests, together with strict process inspection, provide complete quality control. All material is identified by lot number and grade, starting with virgin powder and continuing throughout each manufacturing operation to finished product.

This quality control and inspection system is approved by the USAF under MIL-Q-5923. The company has stated that an affidavit attesting to conformance with specification AMS-3651 polytetrafluoroethylene will accompany every shipment of "electrical grade" Fluoroflex-T products manufactured from Teflon powder. Certified test reports of the actual properties of any shipment will be furnished whenever requested.

Copper supply in 1952 continued to be inadequate for civilian and defense requirements, according to the Bureau of Mines, U. S. Department of the Interior. Actually slightly less copper was available in 1952 than in 1951.

In May of 1952 when copper exports from Chile were embargoed, the President authorized release of 22,000 tons from the strategic stockpile to meet the temporary emergency. By August the situation had improved sufficiently to allow the Defense Production Administration to move copper from the list of most critical materals to the list of those in approximate balance.

JET CUTAWAY: Security regulations have been relaxed to permit the first cutaway view of the Fairchield J-44 jet engine to be published. Currently in production for target drone and missile applications, the engine weighs 300 pounds and is in the 1000 pound thrust class. Length is 72 inches; diameter is 22 inches



Industrial Designer Says Sports Car Styling Not For U.S.

The auto industry's current turmoil over foreign styling for American cars has produced at least one dissenting opinion. Carl W. Sundberg, head of the industrial design firm of Sundberg-Ferar Inc., said recently, "Some car makers are optimistically trying to put a Continental-style dreamboat in every garage by emphasizing the design of a few custom built European sports cars. They have featured Continental styling and have retained European designers, hoping to get an edge on their competitors. I believe this decision will be proved unsound.

"Digging up long-buried gimmicks of the hoary past, such as wire wheels and exposed tire mounts, is only a feeble approach to the problem of improving car design. This copy-cat approach is retrogression of the worst sort."

Although many automobile manufacturers have built experimental cars incorporated Continental styling and some are contemplating production of such automobiles, the following statements indicate Sundberg feels that present American cars are well suited to their function.

"The mass-produced European car, unlike its custom-built brother, is just a crude imitation of the cheapest American models. It fails to meet even minimum American standards for riding comfort, maneuverability and safety. A prime example is the German Volkswagen which looks like a surly bulldog and rides about as well.

"The auto industry should never lose sight of the basic requirements for a saleable American car. It is essentially a 'family bus' which serves as transportation for

SKY-PARK: Vertical double parking machine lifts two parked cars in the air to make room for two more underneath. Hydraulically operated, the device manufactured by Simmons Machine Tool Corp. may be used in garages with 12-foot ceilings. Lifting and securing of the two cars is said to require only 30 seconds

work, for sightseeing and travel, as taxi, school jitney and grocery truck, and must also be the last word in style. Sacrificing any of its versatility and comfort by giving it the appearance of a Continental-type sports car which was designed primarily as a plaything rather than an all-around transportation, is likely to prove disastrous."

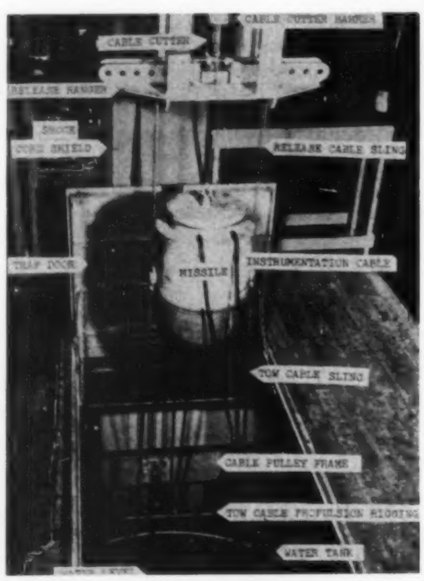
A new division of The Yale & Towne Mfg. Co.—the Powdered Metal Products Division—has been established, consolidating the company's operations in the field of powder metallurgy. The new division unifies under one management the recently acquired Powdered Metal Products Corp. of America, Franklin Park, Ill., and the American Sintered Alloys Division, Bethel, Conn., acquired by Yale & Towne in 1952.

George L. Bachner, formerly president of PMPCA, has been appointed general manager of the new division. Identified with powder metallurgy for fifteen years, Mr. Bachner is a director of the American Society of Tool Engineers, a member of the American Ordnance Association, and a consultant to the Chief of Ordnance, U. S. Army, on powdered metal uses in ordnance components.

Drop Testing Facilitated By Freight Elevator and Shaft

A hole in the floor of a freight elevator and a tank of water at the bottom of the elevator shaft, plus a release system and instrumentation, combine to form a drop test facility at the Mechanical Evaluation Division of the Naval Ordnance Lab. Information on water entry shock problems is obtained by propelling models and missiles into the tank of water at velocities to 150 fps.

The 9-foot diameter tank is 12 feet high and weighs 6 tons. With its contents of 10 tons of water and 15 tons of sand, the tank is supported by a 4-ton platform resting on 165 auto springs weighing approximately 3 tons. A unique feature of the tank instal-



Freight elevator and shaft converted to drop test facility may someday be just an elevator again

lation allows the water surface to be inclined to as much as 45 degrees to the horizontal. This seemingly magical feat is performed with the assistance of a thin plastic membrane which retains the water in the desired position but does not affect shock entry characteristics of the missile being studied.

Missiles or models being tested are actually propelled, rather than dropped from the elevator, by a shockcord sling shot. Missile velocity is regulated by the amount the sling shot is stretched before releasing the missile; raising or lowering the complete elevator platform varies sling shot stretch. The sling is released by cutting the cable used to hold the sling. This propelling system is said to hit the mark every time and also allows control of both pitch and yaw angles. High fidelity instrumentation attached by cable to the missile records data from 6 to 12 accelerometers mounted in various positions.

The Sheffield Corp. will design, manufacture and market Cavitron machine tools, according to a joint announcement by Cavitron Corp. and Sheffield. The agreement provides that the Cavitron Corp. will continue to produce the ultrasonic transducers and generators, being the originators of the ultrasonic machining process and holders of the basic patent rights, and will issue licenses for the process through Sheffield.

The Cavitron machine tool utilizes an ultrasonic vibrating head and an inexpensive tool (such as soft steel) for the precision cutting of hard and brittle materials, either ferrous or nonferrous, such as the carbides, germanium, hardened tool steels, sintered aluminum oxide, quartz, industrial crystals, ferrites, various types of glass, ceramics, sapphire, and other materials difficult to machine. It may be used for precise external or internal machining similar to cutting, drilling and grinding of holes, cavities, slots and recesses of any shape, as well as precision forming of external profiles. Fine surface finish is said to be obtained, and there is no local heating of the work surface or chemical or physical change in the work piece. High cutting

speeds and precision are attained by the use of an ultrasonic tool head vibrating from 18,000 to 30,000 cycles per second.

Restyled Motors Meet NEMA Standards

A new line of electric motors in open drip-proof, totally-enclosed fan-cooled and explosion-proof enclosures meets latest National Electrical Manufacturers Association frame size standards, and features new, modern styling by Sundberg-Ferar, industrial designers. Smaller and lighter than previous motors of equal horsepower, the new L. A. line of Louis

New open, drip-proof (above) and totally-enclosed, fan-cooled motors (below) meet latest NEMA standards. These and other types meeting the new standards will be available in 1954

Allis Co. motors will be available in ac polyphase types in ratings to 40 hp at 3600 rpm in January, 1954.

Despite reduced size and weight of the new motors, performance is equal to that of previous motors. Starting and breakdown torque, efficiency, power factor, temperature rise and service factors have not been sacrified. Improved ventilation and the elimination of unnecessary material make this possible.

Improved bearing construction has also been incorporated to give longer life and better overload capacities for V-belt drives. Factory lubricated, the bearings should give years of satisfactory service without maintenance or attention, the manufacturer claims.

"He never comes right out and says anything."

A private seminar on industrial standardization will be held from January 25 through 29, 1954, in the Engineering Societies Building, 29 West 39th St., New York. The Gaillard Seminars were started in 1947 upon request from companies for assistance in the organization of their standardization work and the training of their men in writing standard specifications. More than 120 organizations have thus far been represented.

The January 1954 session will consist of ten conferences, one each morning and afternoon, Monday through Friday. At each conference a subject on the seminar program will be presented by Dr. Gaillard followed by a roundtable discussion. Further details concerning registration may be obtained by writing Dr. John Gaillard, 400 West 118th St., New York 27, N. Y. or the American Standards Association, 70 East 45th St., New York 18, N. Y.

Community Aids Engineer Development

A community project to assist in the professional development of young engineers was recently described by Cornelius Wandmacher, professor of civil engineering at the University of Cincinnati, at a joint meeting of the American Society for Engineering Education and the Engineers' Council for Professional Development.

The program, started in Cincinnati, embraces six basic points of endeavor: orientation and training in industry, continued education, integration into the community, professional registration, self appraisal, and recommended reading. Nine of the city's industrial companies, 23 local sections of national technical and scientific societies, and members of the administration of the University are participating in the program.

A total of 285 engineers and scientists have enrolled in advanced courses at the University of Cincinnati. Ground was broken during the past year for a $100,000 addition to the city's Engineering Societies building. Committees made up of local businessmen, professional engineers and educators are working on the program.

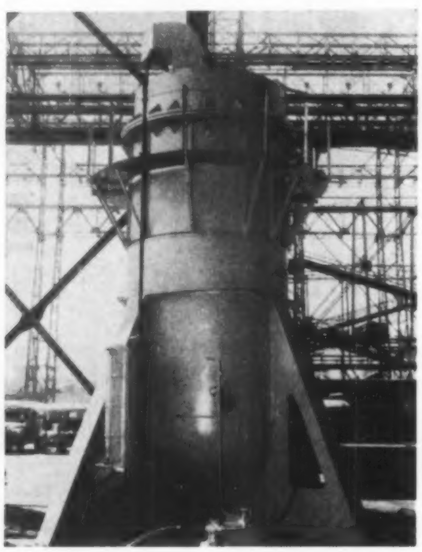
"We invite assistance in expanding this program to other communities in order that the level of professional competence may be raised throughout the nation," Professor Wandmacher, chairman of the ECPD training committee, said.

Thirteen winners of the 1953 Resistance Welder Manufacturers Association Prize Contest have been announced. The RWMA prize contest selects three papers from an industrial source, two from a university source, and one from an undergraduate source. First prize of $750.00 for a paper from an industrial source, entitled "Design

and Application of Edge Ring Welding" was awarded to Chester A. Czohara of International Harvester Co.'s Manufacturing Research Dept. in Chicago. He is a welding engineer conducting research projects in the resistance welding field.

Pressure Chamber Like Space Ship

An unusual pressure chamber for testing models hydrostatically might be mistaken for a space ship of the future. Built by the struc-



Chamber on its tripod supports might be a space ship ready for launching

tural and welding shops at the Norfolk Naval Shipyard, the chamber, 24 feet high, eight feet in diameter and capable of withstanding 1000 psi, has a cylindrical body and hemispherical head.

Pressure chamber head made of cold-formed one-inch thick steel segments

Despite expert opinions that the chamber could not be made, and failure of the first spun head tried, a head was successfully fabricated of cold-shaped steel sections welded into hemispherical form. A bumping process was used to form the segments in a standard eight-inch hemispherical bumping die. After forming, segments were carefully cut to finished dimensions, fitted together and welded.

Lab Corrosion Tests Match Field Test Results

Results of laboratory corrosion tests conducted by the National Bureau of Standards on ferrous metals in 15 different types of soils show good correlation with the actual corrosion observed on wrought specimens in long term tests. Reliable data by the new test method can be obtained in six months while field tests lasting 10 years or more are usually required.

Corrosion of metals in many soils is attributed chiefly to differences in the accessibility of oxygen to the surface. Differences in oxygen concentration create differences in the potential of various areas, resulting in the flow of electric current through the soil from the oxygen-deficient areas (anodes) to those areas (cathodes) receiving more oxygen.

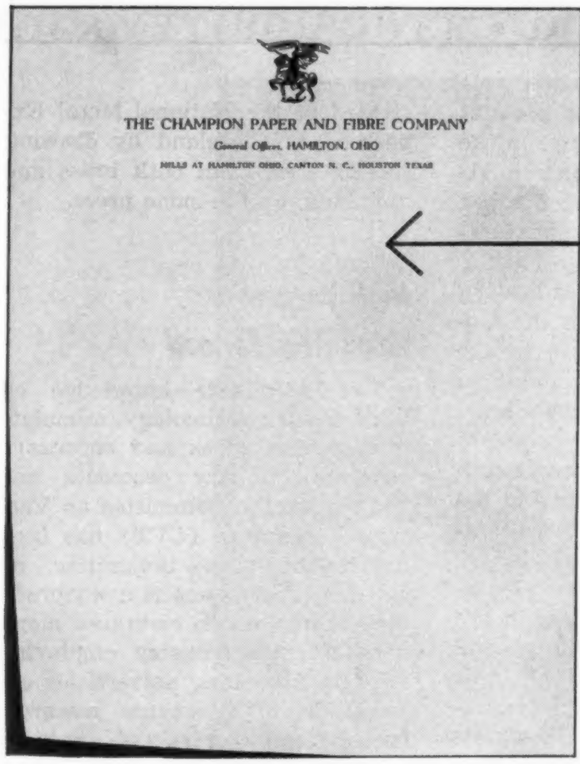
Laboratory corrosion cells used by NBS make use of this principle. Each cell consists of a cylindrical Lucite enclosure containing two short-circuited disk-shaped electrodes of the same material, iron or steel. One electrode is solid, the other perforated, and they are separated by a layer of moist soil. Edges and one face of each of the two electrodes are covered with bituminous paint. The layer of moist soil in contact with the coated side of the perforated electrode regulates diffusion of air to the electrodes; the perforated electrode receives a greater supply of oxygen than the solid one, adjacent to which the soil was puddled, and therefore assumes a more positive potential.

If the corrosion of ferrous metals in all soils proceeded at a rate that was constant with time, weight-loss measurements made after a short arbitrary period of exposure should indicate relative rates of corrosion, assuming negligible cleaning errors. Actually, however, corrosion in soils may proceed at a rate proportional to time, or may sometimes cease entirely, depending on the type of soil. It was thus necessary to operate the laboratory corrosion cells for a sufficiently long period to produce effects comparable to those shown by field exposure tests.

In the NBS investigations, the corrosion cells were kept on short circuit for six months. During this time periodic measurements of cell current permitted study of the progress of corrosion with time. At the end of the period the combined weight losses of the two electrodes were taken as the measure of corrosion.

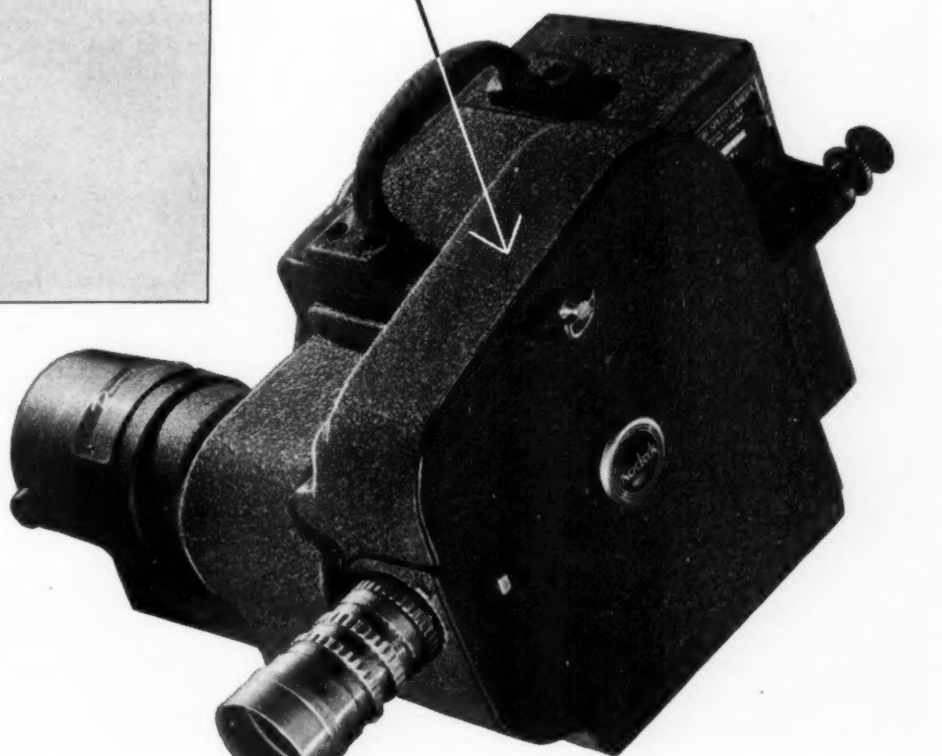
To evaluate the laboratory corrosion cell as a means for predicting corrosion in the field, samples of soil collected at 15 NBS exposure sites were used. The results were correlated with weight-loss measurements made on short lengths of steel and wrought iron pipe which had been exposed for 10

"Some day I want to congratulate the guy who put these indicators up here."


THE CHAMPION PAPER AND FIBRE COMPANY
General Offices, HAMILTON, OHIO
MILLS AT HAMILTON OHIO, CANTON N. C., HOUSTON TEXAS

Everyone knows this is a fine brand of paper

This camera helps make it so



Sometimes problems involving fast-moving mechanical action or fluid flow are more quickly solved by taking the "long look" that high speed movies give you.

Champion Paper and Fibre Company engineers do this with a Kodak High Speed Camera. By filming a troublesome operation or process at 3200 pictures a second and then projecting the films at normal speed on a standard projector, action appears to slow down as much as 200 times. This often makes it possible to see exactly where the root of the trouble lies, and the films can be shown over and over again, stopped at critical points for detailed study and analysis.

With its Kodak High Speed Camera, Champion has studied the factors involved in the formation of a sheet of paper from a pulp suspension, investigated tests used to evaluate paper, studied mechanical problems connected with mill machinery.

Perhaps you have a high speed problem that has to be seen to be solved. Perhaps you find that "cut-and-try" experimentation is too time-consuming and costly. A Kodak High Speed Camera may be your answer. It has the right speed range for most industrial applications, is not too difficult to operate, uses regular 16mm movie film. To see how others have used it, send for our new booklet, "High Speed Motion Picture Making in Industry." Or write for details on a sound movie, "Magnifying Time."

Industrial Photographic Division
EASTMAN KODAK COMPANY, Rochester 4, N.Y.

the Kodak *HIGH SPEED* Camera

Kodak

years at the same 15 localities. An empirical equation relating the laboratory and field data indicated that the laboratory measurements could be used to predict field weight losses with root-mean-square error of about ± 4 ounces per square foot for corrosion rates ranging from 2.5 to 30 ounces per square foot for the 10 year exposure period. Because the 15 soils covered the range of corrosivity of the soils of the United States, this empirical relationship may be applicable to any soil.

Some underground structures, such as piping systems and storage tanks, usually fail as a result of pitting, this factor was also considered. Analysis of the field data showed a good correlation between weight losses and maximum pit depths on wrought ferrous specimens which had been exposed for 10 years at 58 NBS exposure sites. These data were then combined with the first empirical equation to obtain a second equation which can be used to predict, with an average error of ± 18 mils, the depth of the deepest pits occurring on wrought iron or steel buried for 10 years. For the prediction of weight losses and maximum pit depths on areas greater than the area of the field specimens and for periods of exposure other than 10 years, use was made of general equations relating weight losses and maximum pit depths with length of exposure and exposed area. Average values for the constants to be used in these equations were obtained.

Vacuum Gage

A new molecular vacuum gage operates in a manner similar to the fairly well known fluid coupling. Turbine and runner of the vacuum gage are concentric vaned cylinders of thin brass stock. Rotated at 3600 rpm by a small synchronous motor, the driving member imparts momentum to the molecules of gas present in the sealed portion of the gage which is connected to the vacuum system. Momentum of the gas molecules impinging upon the blades of the other cylinder cause it to rotate some amount proportional to the pressure in the system. The driven cylinder is attached to a magnetically damped pointer, and prevented from rotating freely by a spring.

As deflection of the indicating needle is dependent upon the molecular weight of the gas in the system, the gage is available in two types, identical except for scale marking. Both cover a range of 0 to 20 mm of dry air; one is calibrated directly for dry air and the other has a linear scale for calibration for other gases.

Expected uses of this new General Electric development are in the fields of vacuum metallurgy and purification, refrigeration unit manufacturing, electron tube manufacturing, vacuum coating and plating and many other processes. Size of the unit is 4 5/16 inches square by 5⅞ inches deep.

Aluminum strip with drawing qualities similar to those of continuous-cast brass strip made by the same firm is being produced by Scovill Mfg. Co. An innovation introduced in the new mill used to produce the strip is the use of annealing furnaces with a nitrogen atmosphere to minimize formation of oxides on the strip. Drawing qualities of the strip were demonstrated at the National Metal Exposition in Cleveland by drawing similar cups from both brass and aluminum on the same press.

"It's that purist from Machine Design!"

High Vacuum Committee Formed

To disseminate knowledge on high vacuum technology, stimulate exchange of ideas and encourage research on new processes and equipment, the Committee on Vacuum Techniques (CVT) has been established. The Committee on Vacuum Techniques is a nonprofit corporation which embraces membership from industry employing vacuum processes, universities engaged in high vacuum research, and manufacturers of vacuum equipment and components.

At a meeting held in New York city in June, some 60 representatives of industries, universities and equipment manufacturers discussed the nature of an organization which would best serve their needs. Dr. Henry Barton of the American Institute of Physics discussed various problems associated with formation of a formalized technical society. He suggested that the group should be kept on an informal basis and concentrate on service to the vacuum field rather than on a formal permanent organization.

A strong opinion was advanced that one of the most useful functions of such an organization would be the standardization of nomenclature, testing techniques, and equipment performance ratings. Considerable confusion has resulted from misunderstanding of such fundamentals since high vacuum is employed by a broad variety of fields such as electronics, food, metallurgy, coating, distillation, pressure vessel testing, optical and nuclear.

The committee on Vacuum Techniques plans to sponsor a High Vacuum Symposium in June, 1954 at Asbury Park, N. J. Programming plans are well underway. The Committee on Vacuum Techniques, Box 1282, Boston 9, Mass., is soliciting membership applications from individuals and companies interested in high vacuum technology.

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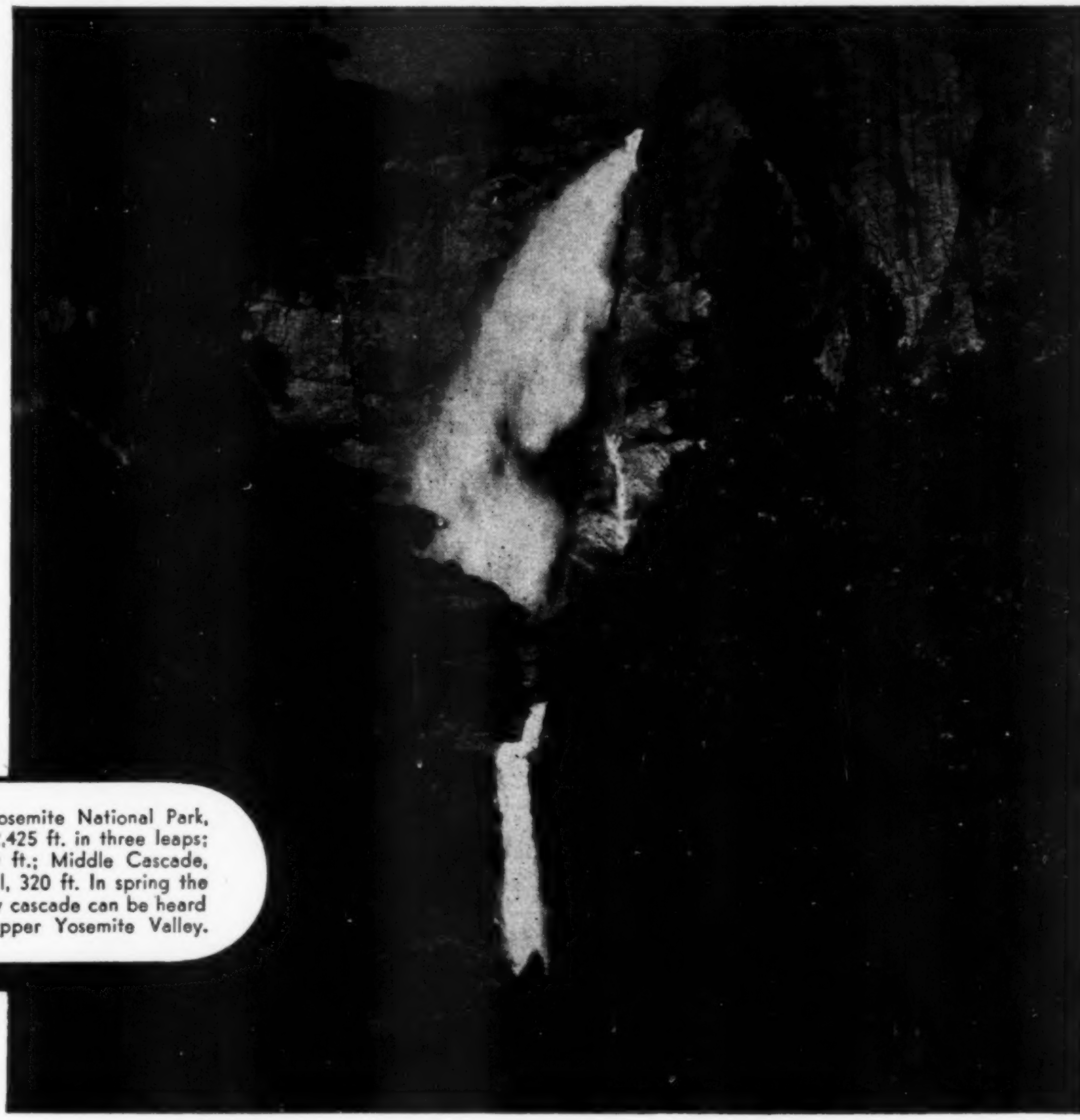

Yosemite Falls, Yosemite National Park, drops a total of 2,425 ft. in three leaps; Upper Fall, 1,430 ft.; Middle Cascade, 600 ft.; Lower Fall, 320 ft. In spring the roar of this mighty cascade can be heard throughout the upper Yosemite Valley.

LOGAN AIR CONTROL VALVES

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- SELF-CLEANING

Model 6245—4-Way, 2-Position

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EFFORT • MOTION

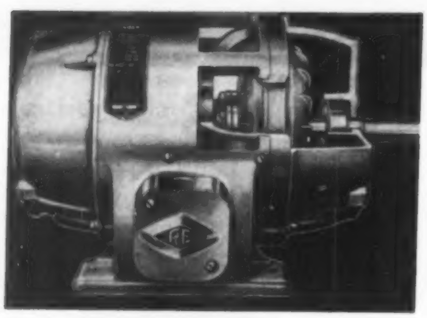
LOGANSPORT MACHINE CO., INC., 811 CENTER AVE., LOGANSPORT, IND.

NEW PARTS AND MATERIALS

For additional information on these new developments, see Page 269

Compact Motors

Newly designed Fluid-Shaft motors use an internal fluid coupling to give greater compactness in terms of output. Unit illustrated is a ½-hp type in a No. 203 frame. Its overall length has been shortened 4 in., its height 1¼ in. and floor to shaft centerline distance ½-in. from previous model. Standard frame sizes from 203 through 326 and ratings from ½ through 15 hp are available. In addition to providing compactness, coupling also gives smooth load acceleration

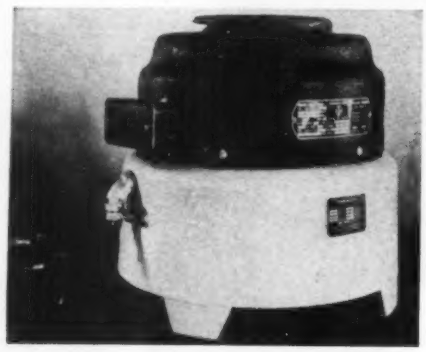


and protection for driven equipment. Made by **Reuland Electric Co.,** 3001 West Mission Rd., Alhambra, Calif.

For more data circle MD-72, Page 269

Frequency Converter

Nobrush compact motor-generator unit converts 60-cycle current to 400 cycles. It has neither brushes, slip rings, commutator nor exciter. It is not affected by temper-

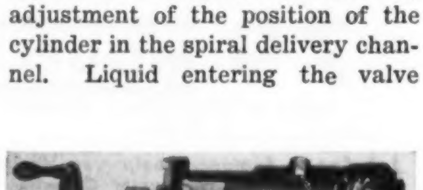


ature extremes, moisture, grit, overloads or short circuits. Outputs of 150 va to 25 kva single or three phase are available with motor drive for any standard frequency or voltage. The two-bearing unit measures 18 in. in diameter, is 16 in. high and weighs 247 lb. Made by **Georator Corp.,** Manassas, Va.

For more data circle MD-73, Page 269

Flow Control Valve

Volume of delivery of viscous liquids is accurately controlled by adjustment of the position of the cylinder in the spiral delivery channel. Liquid entering the valve

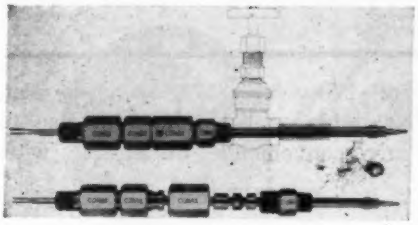


housing from either of two ports is forced into the spiral channel on the inside wall of the valve through which the liquid escapes. This channel is nearly 3 ft long and about the size of ⅜-in. pipe. As the wheel control is turned, the cylinder exposes more of the spiral channel, and the length of flow is reduced. Flow is controlled by friction on the liquid rather than by the size of the opening. Handling paints, crude oil and waste liquids containing dirt or sediment without clogging, the valve is available in 1¼, 2 and 2½-in. sizes from **Anglo American Mill Sales Co.,** Owensboro, Ky.

For more data circle MD-74, Page 269

Thermocouple Packing Gland

C-U-P gland permits changing exposed wire thermocouples without reducing pressure on the system. It is basically a thermocouple gland equipped with an open end protection tube, a special packing gland and a gate valve. The protection tube enters the pressure system through the packing gland and gate valve; the thermocouple gland provides the pressure seal around the exposed wires inside the tube; and the packing gland seals the outside of the tube in normal operation and while the thermocouple and tube are being slid in or out of the system. When the thermocouple is pulled back past the gate of the valve, the valve can be closed to hold the pressure in the system. Complete removal of the thermocouple is then accomplished by loosening the packing gland cap

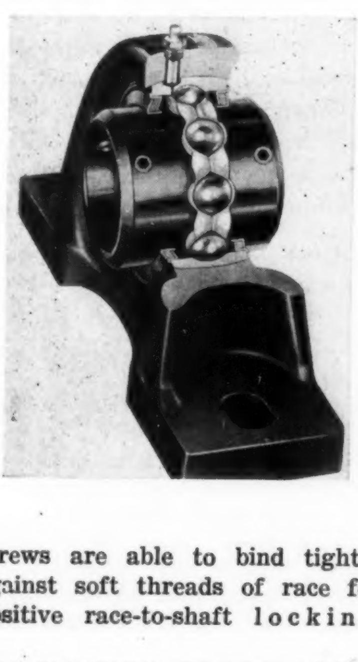


and pulling out the gland follower, sealant and removable gland seat. A stainless steel setscrew seated in a machined groove in the thermocouple gland prevents blowback of the tube through the packing gland. The C-U-P gland normally withstands pressure of about 500 psi. Temperature range is determined by the sealant selected for both the thermocouple gland and the packing gland parts. Asbestos-graphite, standard for the packing gland, will withstand 700 F; Teflon, 500 F; Neoprene, 200 F. Available for use with 14 or 20 gage thermocouple wires from **Conax Corp.**, 4515 Main St., Buffalo 21, N. Y.

For more data circle MD-75, Page 269

Pillow Blocks

Manufacture of Sealmaster ball bearing units with double extended race embodies Zone Hardening, a process for induction heat treating the inner race ring through the ball path section only. Since extended portions of race remain unaffected, hardened threads of set

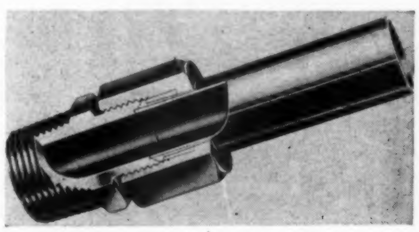


screws are able to bind tightly against soft threads of race for positive race-to-shaft locking. Double extension of inner race ring assures positive lock between inner race of bearing and shaft and reduces shaft wear and fretting corrosion. Shock and vibration resistance are increased since most of inner race remains relatively soft and tough. Zone Hardening is done by automatic induction heating and high-speed oil quenching. The bearings are made by **Stephens-Adamson Mfg. Co.**, Aurora, Ill.

For more data circle MD-76, Page 269

Flareless Fitting

Steel, brass, aluminum and stainless steel flareless fittings are made in all popular shapes in sizes from 1/8 to 2 in. They are suitable for heavy or thin wall tubing, neither of which is cut or gouged when fittings are tightened. Cap-



able of handling anything that flows, fittings hold twice tube bursting pressures up to 50,000 psi, depending on size of fitting used. Performance exceeds that required by JIC hydraulic standards and AN-F-47 and MIL-F-5506. Fittings can be reused by replacing a sleeve. Made by **Aircraft Fitting Co.**, 1400 E. 30th St., Cleveland 14, O.

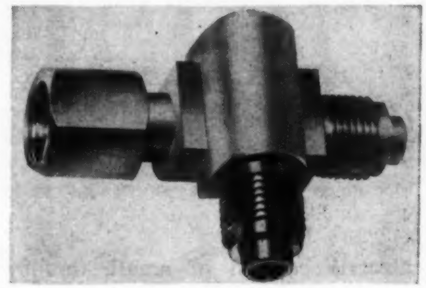
For more data circle MD-77, Page 269

Miniature Metal Seals

Tourseals are available made from 1/32-in. diameter stainless steel tubing with 0.005-in. wall thickness. Sizes range from 3/8 to 1 in. OD. Seals can be used at temperatures ranging from −150 to 1000 F. They seal against pressures of 1000 psi and are resistant to corrosion. Used in small hydraulic, pneumatic and steam lines, the seals are adaptable to bolt heads, flanges, rivets, etc. Special sizes can be made to order by **D. S. D. Mfg. Co.**, 2964 Whitney Ave., Hamden, Conn.

For more data circle MD-78, Page 269

Dual Drive Adaptor



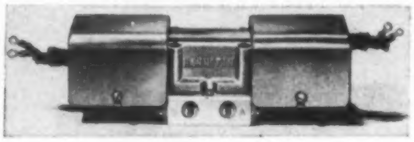
Dual drive, Circle Ess adaptor is designed for use with flexible shaft drives where more than one unit is to be driven from the same source or where space limitations prevent direct connection without bending shaft to point of possible damage. Unit has 5/8-18 threaded connections with 0.104-in. internal square drives. Ratios of 1:1, 15:1, 16:1 and 17:1 are available for either left or right-hand rotation. Body is aluminum alloy with brass connectors. Gears are hardened steel and permanently lubricated for high or low temperature service. Weight is 4 oz. Made by **F. W. Stewart Mfg. Corp.**, 4311 Ravenswood Ave., Chicago 13, Ill.

For more data circle MD-79, Page 269

Solenoid Valves

For speeds as high as 600 cycles per minute, two 1/4-in., four-way poppet valves are direct solenoid operated. Model CC1-25 is a single-solenoid model; its single valve stem, which is the only moving part, is spring returned. Model CC11-25 has two solenoids, and its single valve stem remains at either end of its short stroke until the other solenoid is momentarily energized to return it. Thus "momentary contact" operation is possible, eliminating the necessity of keeping the solenoid energized to hold the valve in its shifted position. Pipe or tubing connections are to a separate manifold, to which the valve mounts. Internal passages of the valves are as large

as or larger than the inside diameter of standard weight pipe of their rated size, and they are offered, optionally, tapped for ⅜-in. pipe or tubing connections. Principal use is expected to be for



electric control of small, double-acting air cylinders on short stroke, high-speed applications such as switching mechanisms for conveyors, welder cylinders and clamping and work-positioning cylinders on various automatic and semiautomatic machines. Made by **Hannifin Corp.**, 1116 S. Kilbourn Ave., Chicago 24, Ill.

For more data circle MD-80, Page 269

Corrosion Preventive

Barcote No. 600 is applied by either brushing or spraying. A solvent, cut-back, leaded compound, it provides coverage with a thin, rubber-like film which sets to a semihard coating of pure lubricant. It adheres to and penetrates all metals. The film expands and contracts without flaking, cracking, pulling away, blistering or failing. The material flows at 0 F, does not emulsify with water, and penetrates behind scale and through rust. It does not evaporate in storage, does not dry hard, resists the action of salt water, is soluble in petroleum solvents, does not clog spray equipment, and can be filtered through an 0.005-in. mesh unit. Developed by the **Brooks Oil Co.**, 934 Ridge Ave., Pittsburgh 12, Pa.

For more data circle MD-81, Page 269

Snap-Action Limit Switch

Features of Loxswitch single-pole double-throw limit switch include a trigger-action break independent of operating lever speed, operating shaft supported by bearings at both ends and large wiring compartment separated from moving parts. Vibration and impact shock are minimized by use of lightweight silver contacts in balanced rotor. Each pole is independently spring loaded and locked when in either position.

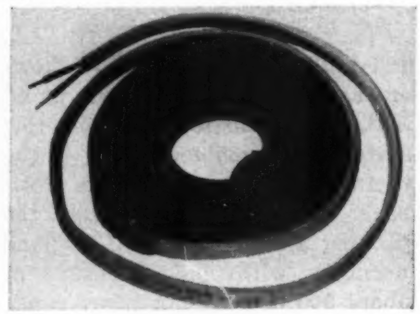
Normally assembled for counterclockwise operation with spring return, switch, if specified, can be assembled for clockwise operation, or with return spring omitted, for maintained contact in either direction. It can be changed in field to meet these conditions. Usable on up to 600-v circuits, switch is rated

6 amp on 120-v ac. With 20 per cent over-travel, units have tested at 20 million operations. Made by **R. B. Denison Mfg. Co.**, 4220 Hough Ave., Cleveland 3, O.

For more data circle MD-82, Page 269

Heating Tape

A wire heating element is sealed in Ribbon Heat flexible plastic tape, one end of which, in the Underwriter approved type, has flexible leads which connect to any 110 or 220-v approved outlet box. The tape is rated at 7 w per foot. Shockproof, waterproof and unaffected by mild acids and alkalies, it is furnished in lengths of 20 to 60 ft for 110-v service and 40 to 120 ft for 220-v use. The tape is also available with a plug, for use in any convenience outlet, or with

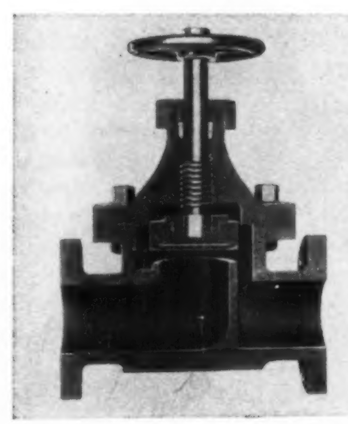


thermostats which provide automatic protection against sudden temperature changes. In original equipment Ribbon Heat is employed to prevent marked differences in temperature between sections of a unit. Developed by **Cox and Co.**, 115 East 23rd St., New York 10, N. Y.

For more data circle MD-83, Page 269

Corrosion Resistant Valves

Made in both natural and Buna N hard rubber, globe, angle and wye valves are available in sizes from 1 to 4 in. All stems and disks are metal reinforced. Disks with soft rubber seating surfaces are renewable. Natural hard rubber valves can be used at temperatures not exceeding 120 F. Line is also available in special compounds



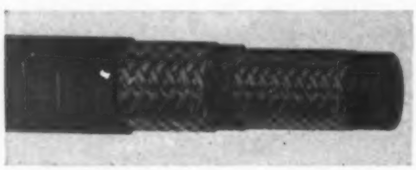
which may be used to 180 F unsupported, as well as in Buna N hard rubber, which is recommended

for service up to 225 F and resists many organic chemicals in addition to the inorganic acids, alkalies and salts to which hard rubber is resistant. Made by **Vanton Pump & Equipment Corp.**, Empire State Bldg., New York 1, N. Y.

For more data circle MD-84, Page 269

Leakproof Flexible Hose

Synthetic tube incorporated in Qua-Seal leakproof flexible hose prevents permeation of highly volatile freon gas. Hose is reinforced with two layers of braided textile and withstands wide temperature range found on both suction and discharge sides of refrigerator compressor. It can be used with working pressures up to 400 psi. Hose can also carry lacquers, lacquer solvents, naphtha, kerosene

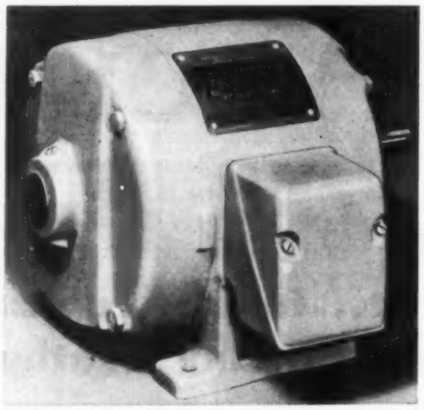


and many chemicals. It is available in inside diameters of ¼, ⅜, ½, ¾ and 1 in. Made by Quaker Rubber Corp., division of **H. K. Porter Co. Inc.**, Tacony and Comly Sts., Philadelphia 24, Pa.

For more data circle MD-85, Page 269

Polyphase Motors

Designated Tri-Clad 55, line of polyphase ac motors has average size reduction of 50 per cent by volume and averages 22 per cent less weight per horsepower than previous models. At the same time line retains cast iron construction and incorporates new insulation system, bearing assembly and ventilation plan. Noise levels have also been reduced so that the new 10-hp motor tests as low as the former 2-hp model and all units pass a quality control sound test. Units will be available in NEMA frame sizes 182 and 184 (1, 1½, and 2 hp at 1800 rpm) in horizontal drip-proof and totally enclosed fan-cooled models, and a complete

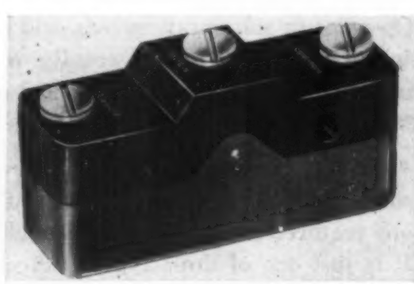


line of gearmotors. Larger frame sizes and other types are planned for production and will be made available at regular intervals. Made by **General Electric Co.**, Schenectady 5, N. Y.

For more data circle MD-86, Page 269

Snap-Acting Switches

New base design of Unimax snap-acting switches provides terminals which are sturdy enough for wiring circuits handling the 17 amperes at which the switches are Underwriter approved. The termin-

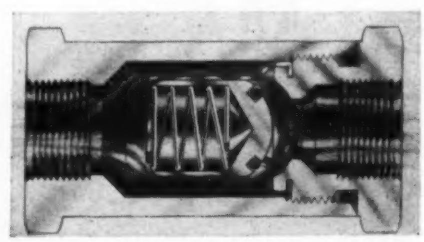


als are uniformly spaced along the switch base, on which a central step lengthens creep distances so that no insulating barriers are required and circuit separation is easily maintained. The flat surfaces and 6-32 binding head screws of the terminals allow connection by standard size lugs or looped wires up to No. 14. Markings on the base identify the normally open, normally closed and common contacts. Designated -5, the base is available on all switches except the bottom-reset and double-throw top-reset types from Unimax Switch Div., **W. L. Maxson Corp.**, 460 W. 34th St., New York 1, N. Y.

For more data circle MD-87, Page 269

Check Valve

Features of this free-flow check valve include low pressure drop and virtually no flow restriction in nonsurge hydraulic systems operating up to 3000 psi. Spherical poppet nose and matching seat are self aligning assuring a leakproof seal. Standard opening pressure is as low as 3 oz, but stronger springs are available so that valve can be used as combination in-line relief-check valve. Various metals



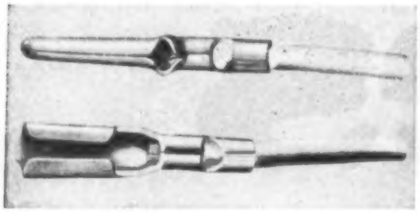
are offered in sizes from ⅛ to 1 in. with all combinations of tube and pipe ports. Operating temperatures are −65 to 200 F, higher on order. Made by **Republic Mfg. Co.**, 1930 W. 77th St., Cleveland 2, O.

For more data circle MD-88, Page 269

Miniature Electrical Connectors

Taper technique eliminates terminal screws or soldering in electrical circuits. Both self-locking terminals, the taper pin with mating receptacle and the flat taper tab and taper tab receptacle were designed for limited space. Taper Pins are only a few thousandths of an inch thicker than the insulated diameter of the wire, which permits very close spacing of terminations without danger of short circuits. The units resist vibration and are tandem mounted in strip

form and wound in reels for automatic application to wire. Typical uses are in telephone mainframes,

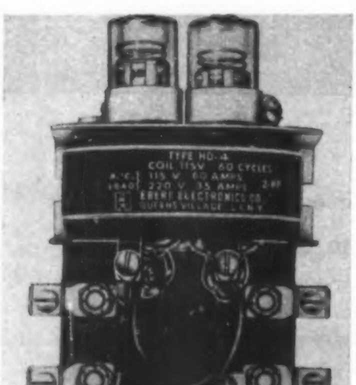


stepping switches and relays, outdoor cable terminals, "AN" connectors in aircraft, and relays, switches and socket tabs in business machines. Made by Electronics Div., **Aircraft-Marine Products Inc.**, 2100 Paxton St., Harrisburg, Pa.

For more data circle MD-89, Page 269

Mercury Contactor

Heavy-duty model HD-4 features mercury-to-mercury contacts; silent, chatter-free operation and hermetically sealed, explosion-proof construction. Each tube of this two-tube contactor will control 60 amp at 115 v ac or 35 amp at 230 v ac. For dc operation, control capability per tube is 12 amp at

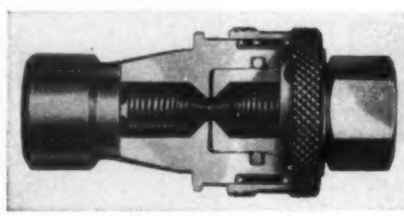


115 v. For motor control, the unit is rated at 2 hp at 115 v ac, single phase; 3 hp at 230/440 v ac, single phase; and ½ hp at 120/220 v dc. Coils are available for various control voltages, and the unit is available with both tubes normally open or both normally closed, or with one tube normally open and one normally closed. Overall dimensions are 5 x 3⅜ x 2¼ in. Made by **Ebert Electronics Co.**, 212-26 Jamaica Ave., Queens Village 28, L. I., N. Y.

For more data circle MD-90, Page 269

Shut-Off Coupling

Hydraulic and pneumatic lines in ⅛ to ⅜-in. sizes can be quickly and easily connected or disconnected with this two-way shut-off coupling which supplements series HK line. Both socket and plug ends of line are instantly sealed when coupling is disconnected by

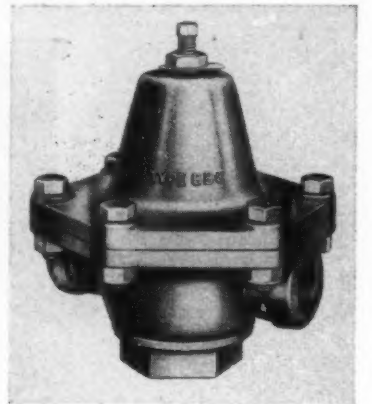


torpedo valves which contact valve seats. When coupling is connected, fluid flows through freely with positive seal against leakage insured by compression of O-ring in socket body against outside surface of inserted plug. All coupling sizes are available in brass or steel. Made by **Hansen Mfg. Co.**, 4031 W. 150th St., Cleveland 11, O.

For more data circle MD-91, Page 269

Pressure Reducing Valve

Intended for service with heavy oil as well as dirty or high viscosity liquids, Cash-Acme type BBC pressure reducing and regulating valve automatically reduces and maintains flow pressure. In cadmium plated iron model maximum reduction is from 200 to 125 psi, while in bronze unit, initial 250 psi is reduced to 200 psi. Direct acting single seated valve has spring-loaded Monel or Neoprene diaphragm and renewable stainless steel piston and seat. Universal joint type seating arrangement insures free valve operation. Square

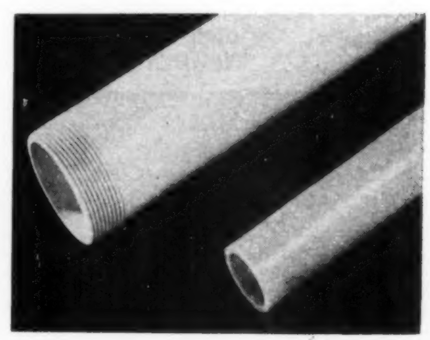


head adjusting screw is standard, while T handle or handwheel are also available. Valve is offered for pipe sizes from ⅜ to 1½ in. Made by **A. W. Cash Valve Mfg. Corp.**, 666 E. Wabash Ave., Decatur 60, Ill.

For more data circle MD-92, Page 269

Plastic Piping

Combining strength with high chemical resistance, Carlon V is an extruded rigid nonplasticized polyvinyl chloride pipe suitable for carrying most corrosive substances. It is practically unaffected by all mineral acids, bases and salts, chlorine, oil, grease,



gasoline, alcohol and carbon tetrachloride, and is completely immune to rot, rust and electrolytic corrosion. It is assembled by conventional means and weighs about half as much as similar aluminum pipe. Also suitable for electrical conduit and insulation, it is available in nominal pipe sizes from ½ to 6 in. with IPS wall thickness schedules 40 and 80. In schedule 80 working

pressures range from 550 psi in the ½-in. size to 195 psi in the 6-in. size. Special extrusions and wall thicknesses are also available to order. Made by **Carlon Products Corp.**, 10225 Meech Ave., Cleveland 5, O.

For more data circle MD-93, Page 269

Running Time Meter

Series C2 running time meters are now available with ranges of 10,000 or 100,000 hours, counting in 1/10 or 1-hour units; and with ranges of 10,000 or 100,000 minutes, counting in 1/10 or 1-minute units. They can be supplied for

115, 220 or 440-v operation at 60, 50 or 25 cycles. Operated by heavy-duty, self-lubricating synchronous motors, the meters are completely enclosed to seal out moisture and dust. Units are housed in 3½-in. black bakelite meter cases and require a 2⅞-in. panel mounting hole. Screw terminals for wiring are located on the back of the unit. Made by **Industrial Timer Corp.**, 117 Edison Place, Newark 2, N. J.

For more data circle MD-94, Page 269

Light Duty Pillow Blocks

Series of light duty type PB ball bearing pillow blocks has housing which provides initial self-alignment in all directions. They are ready to mount and include separable two-piece pressed steel housing, wide inner ring ball bearing with locking collar and bearing seals. Bearing is lubricated for life and is recommended for speeds up to 2400 rpm. Radial load ca-

pacity varies between 300 and 600 lb, depending on shaft size. Standard shaft sizes are from ¾ to 1¼ in. in increments of 1/16 in. Made by **Fafnir Bearing Co.**, New Britain, Conn.

For more data circle MD-95, Page 269

High-Capacity Relief Valves

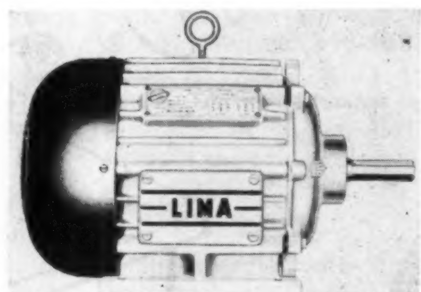
Designed to provide constant relief pressure control in oil systems, 1-in. high-capacity pilot type relief valves incorporate small pilot valves which control the action of larger main valves. Constant relief pressure is not affected by flow, temperature or oil viscosity. The main valve is responsive only to the flow through the pilot and positions itself to maintain this pilot flow constant. Any change in pressure at the pressure point affects the pilot, which in turn controls the main valve. High capacity is obtained by sliding spool construc-

tion. Relief pressures are adjustable from 50 to 1000 psi or 500 to 3000 psi. Capacity is 40 gpm at pressure of 200 psi or greater. Made by **Fluid Controls Inc.**, 1284 N. Center St., Mentor, O.

For more data circle MD-96, Page 269

Totally Enclosed Motors

High velocity movement of air over the deep, integrally cast fins of type E totally-enclosed, fan-cooled motors does not permit dust, dirt or other foreign matter to accumulate on the motor frame.



Motors are made in ratings of ¾ hp at 900 rpm to 20 hp at 3600 rpm in NEMA frame sizes 224 to 326 inclusive. Additional features are rigid cast iron frames with integral feet, double-width prelubricated sealed ball bearings, cast aluminum fan and availability for vertical, wall, ceiling or horizontal mounting. Two or three-phase motors are offered in all standard frequencies and commercial voltages below 600 v. Made by **Lima Electric Motor Co.**, Lima, O.

For more data circle MD-97, Page 269

Heat Resistant Paint

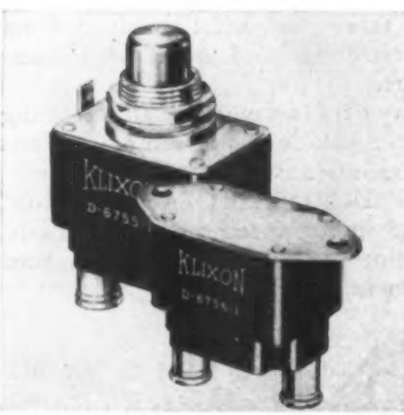
Sicon silicone base paint provides a protective coating which withstands temperatures to 1000 F. It can be applied by brushing, spraying, dipping or roller-coating. Color range includes various shades of tan, brown and gray, as well as yellow, red, green, blue, ivory and white. The finish resists flaking, peeling and discoloring after exposure to heat. Uses

include appliances incorporating heating, and aircraft assemblies. A means of identification of components subject to intense heat is also afforded by the permanent colors of this coating. Made by **Midland Industrial Finishes Co.**, Waukegan, Ill.

For more data circle MD-98, Page 269

Circuit Breakers

Klixon D6755-1 and D6756-1 circuit breakers are completely sealed and small in size. Ratings are $\frac{1}{2}$, $\frac{3}{4}$ and 1 amp. Both the manual reset nonindicating type and the automatic reset version are small and have as their actuating element a snap-acting Spencer disk.



Both withstand shock of over 25 *g*, and vibration frequency of 10 to 55 cycles per second. Made by Spencer Thermostat Div., **Metals & Controls Corp.**, Forest St., Attleboro, Mass.

For more data circle MD-99, Page 269

Midget Solenoid Valves

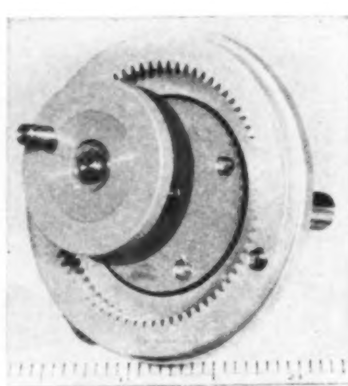
Suitable for Class 1 Group D hazardous locations, explosion-proof solenoids have been added to ASCO midget line. They are available for all commercial and industrial voltages. Enclosures are cast iron, with $\frac{1}{2}$-in. threaded conduit connections. Conduit boxes revolve 360 degrees. Two-way units are available normally-closed for shut-off, adjustable main flow and adjustable by-pass service, and normally open for shut-off duty. Standard three-way universal units

are suitable for any three-way control application, without adjustment and regardless of flow direction or points of pressure application. Available materials are brass and stainless steel, with maximum working pressures for the two-way unit being up to 1000 psi. Made by **Automatic Switch Co.**, 391 Lakeside Ave., Orange, N. J.

For more data circle MD-100, Page 269

Sine-Cosine Mechanism

Accurate conversion of angular rotational movements into linear sine or cosine movements is function of this sine-cosine mechanism. It offers instantaneous solutions to problems of changing variables

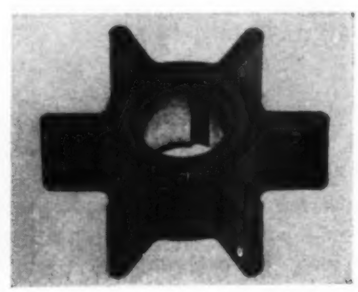


involving vector components, range and bearing compution, and other trigonometric functions. Mechanism has a conventional three-lug arrangement for mounting, weighs 2 oz and is accurate within 0.2-per cent full scale. Stroke is 0.750-in. plus or minus 0.0015-in. each side of zero. Made by **Librascope Inc.**, 1607 Flower St., Glendale 1, Calif.

For more data circle MD-101, Page 269

Rubber Bonding Process

Method of bonding natural, synthetic or silicone rubber to metal permits grinding of parts with metal cores concentric to tolerances of ± 0.002-in. Applications of this process include nuts and bushings which can be tightened without breaking, valves with improved seating even where grit is encountered, synthetic rubber washers which reduce water absorption, and rubber insert backings on hydraulic cups which provide more support than solid rub-



ber. This process makes possible the use of metal in many rubber parts requiring rigidity as well as resiliency. Developed by **Advance Rubber Co. Inc.**, 1702 Washington Ave., N. Minneapolis, Minn.

For more data circle MD-102, Page 269

Cobalt Base Alloy

Haynes Alloy No. 25 (L605) sheet is cold rolled in thicknesses down to 0.0005-in. This alloy has high strength and hardness and excellent corrosion resistance at temperatures up to 1500 to 2000 F. Cold working results in a range of hardness values as high as Rockwell C-55, which is produced with an accompanying ultimate strength of about 322,000 psi and an elonga-

tion of about 2 per cent. This results from 50 per cent cold reduction followed by an aging heat treatment. Available from General Plate Div., **Metals & Controls Corp.**, Attleboro, Mass.

For more data circle MD-103, Page 269

Snap-Action Switch

A contact pressure of at least 45 grams is maintained in this hammer-blow snap-action switch until the contacts snap over. This assures reliability where vibration might otherwise cause switch to operate prematurely. Contacts snap over with hammer-blow force of 150 to 200 grams, reducing arcing and increasing electrical capacity. Rated 15 amp, 125/250 v ac or 30 v dc, switch is single-pole, double-throw type with four terminals and two separate circuits in 1½ x ½ x ½ in. case. It is also available in a reset model.

Switch can be used with or without actuator, depending on overtravel required. Lever, leaf, push-button and toggle actuators are available. Made by **Electro-Snap Switch & Mfg. Co.**, 4220 W. Lake St., Chicago 24, Ill.

For more data circle MD-104, Page 269

Miniature Hydraulic Valve

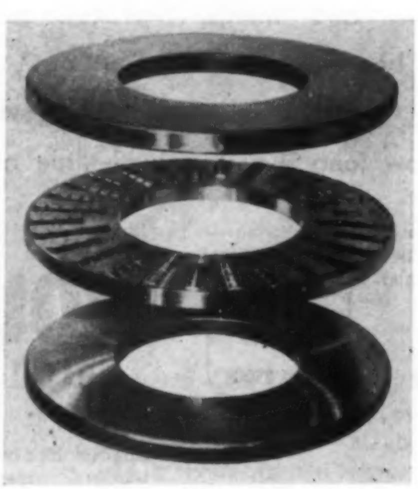
Compact V10-1000 miniature hydraulic valve can be utilized as a two or three-way valve in either normally open or closed positions. Principal applications are in automatic operation of single and double-acting hydraulic cylinders or for pilot control of large three and four-way hydraulic valves. It handles hydraulic fluids at pressures up to 1000 psi. Internal parts are stainless steel. Valve

has ⅛-in. NPT threaded ports, effective orifice diameter of 3/32-in. and measures 3 7/16 in. high by 1⅝-in. diameter. Range of alternating and direct current voltages is available. Made by Skinner Electric Valve Div., **Skinner Chuck Co.**, 95 Edgewood Ave., New Britain, Conn.

For more data circle MD-105, Page 269

Thrust Bearings

Modifications in high capacity T-59 thrust bearing for heavy duty applications on extruders, generators, pulverizers and similar equipment raise its load capacity by about 9 per cent and its life by 29 per cent. New retainer design sets long roller slots of five

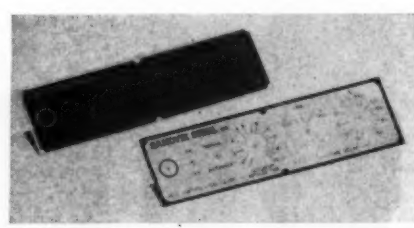


rollers each in alternate positions with short roller slots of four rollers each. Staggered slotting increases number of rollers exposed to the thrust load, thus increasing capacity. Bearing will carry a 590,000-lb load at 100 rpm. Made by **Rollway Bearing Co.**, Seymour St., Syracuse 4, N. Y.

For more data circle MD-106, Page 269

Nameplate Process

Using a photochemical process, ten nameplates can be made at a unit cost almost as economically as a quantity of 500. Nameplates can be supplied on any nonporous surface such as metal or glass. They can also be made of 0.003-in. aluminum foil backed with a high tensile bonding material. Various colors are available. Process can also reproduce 120-line half-tone prints with fine detail. Nameplates can be made from customer's art



work, or rough sketches can be made into the finished art wo by the originator of the process. Company will also supply process and materials to other companies. Developed by **Henry G. Dietz Co.**, 12-16 Astoria Blvd., Long Island City 2, N. Y.

For more data circle MD-107, Page 269

Beryllium-Copper Alloy

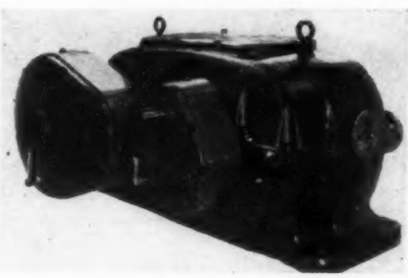
High strength and hardness values are combined with toughness and good electrical conductivity in Beryldur, a beryllium-copper wrought alloy. Properties can be varied at will by simple heat treatment. Beryllium content is 1 per cent. Alloy is produced in two tempers—heat treatable and mill hardened. Well suited

to manufacture of springs and switches, alloy is made in strip form only at the present time but will be produced in wire and rod also. Made by **Beryllium Corp.**, Reading, Pa.

For more data circle MD-108, Page 269

Variable Speed Drives

Positive infinitely variable speed drives for 20 to 25-hp applications, HG-6 and HGG-6, are furnished with either a single reduction input or output helical gear attachment, or with both single reduction input and output attachments respectively. Input gear reductions are available from 1.93 to 1 minimum to 5.82 to 1 maximum and output gear reductions from 1 to 1 to 6.33 to 1. Speed increasing gear



sets are also available. Drive provides accurate positive power control by means of a drive chain with self-forming metal teeth which engage with radial grooves in two pairs of cone-shaped wheels. Ratio is changed by altering effective wheel diameter. Drive is unaffected by extreme temperatures, dust, humidity and explosive vapor conditions. Made by **Link-Belt Co.**, 307 N. Michigan Ave., Chicago 1, Ill.

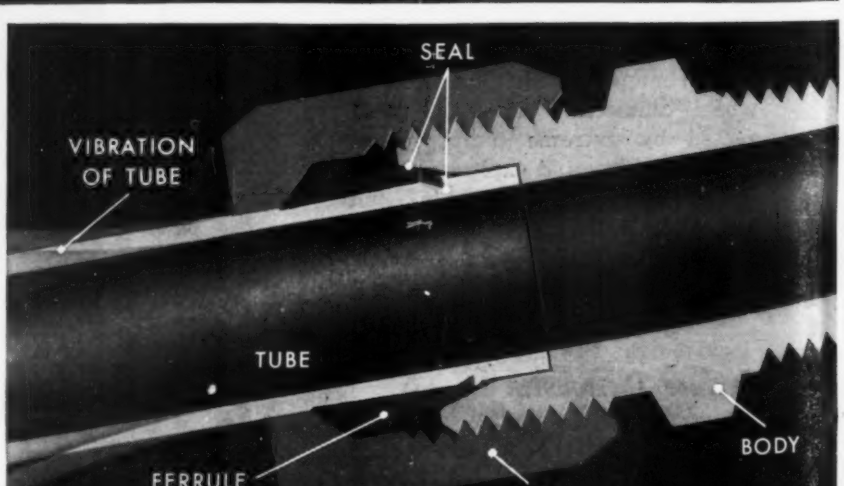
For more data circle MD-109, Page 269

Surface Thermostat

Rigidly constructed unit, with a deep-drawn housing of either brass or steel, withstands mechanical shock. Strip type construction is such that the thermal responding element is insulated from the

(*Continued on Page 292*)

Shaking furiously, this machine, built by Columbia Machine Works of Vancouver, Washington, molds 8 building blocks per minute. It compresses concrete, vibrates it under 350 psi, and ejects finished blocks.



Cutaway shows how vibration is damped by the rear bevel of the ferrule. Note how it grips the tube. This prevents stresses at the double seal and assures a leakproof connection. The nut also helps support the tube.

GOT THE SHAKES?

Solve your tube vibration problems with leakproof Parker *Ferulok* flareless fittings

This machine shakes from stem to stern. Because of the extreme vibration, maintenance of hydraulic lines was a big problem until the design was changed to specify Parker *Ferulok* flareless fittings. Since then the manufacturer has not had a single call for hydraulic line maintenance nor received any customer complaints.

Here's what makes *Ferulok* flareless fittings leakproof on such critical applications:

These flareless steel fittings have a double seal. Wedging action of the ferrule, when drawn down by the nut, forms a seal between body and ferrule. The cutting edge of the ferrule "bites" into the outer surface of the tube, forming a second positive seal. Self-centering action assures an even "bite" around the circumference of the tube. The extent of the bite at the cutting edge of the ferrule is completely visible when the fitting is taken apart.

Vibration won't break these seals. The rear bevel of the ferrule grips the tube. This damps vibration and prevents stresses from concentrating at the line of "bite".

Parker *Ferulok* flareless fittings are designed especially for high-pressure, heavy-wall tubing. *Ferulok are the fittings that conform with the new S.A.E. Hydraulic Flareless Tube Fittings Standard.* Of course, they also meet J.I.C. Hydraulic Standards for flareless fittings.

You have no assembly problems when you specify Parker *Ferulok.* Simply use a wrench; no special tools are needed.

Why not specify Parker *Ferulok* flareless fittings for *your* heavy-wall tubing applications? We now offer a greatly expanded *Ferulok* line with a complete range of body shapes and sizes for tubing through 2 inches, outside diameter.

Tube and Hose Fittings Division
The Parker Appliance Company
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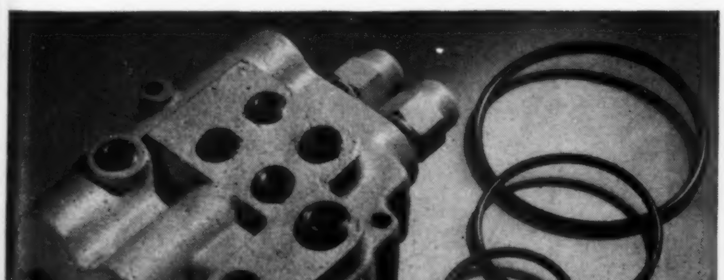
Parker
Hydraulic and fluid
system components



What other Parker products for hydraulic and fluid systems interest you? Triple-lok flare fittings? Hydraulic control valves? Synthetic rubber O-rings? Any other Parker products?

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Send for catalog of complete information about Parker tube fittings. If you'd like to know about Parker products, please list them on the coupon or write to the above address.

New Parts

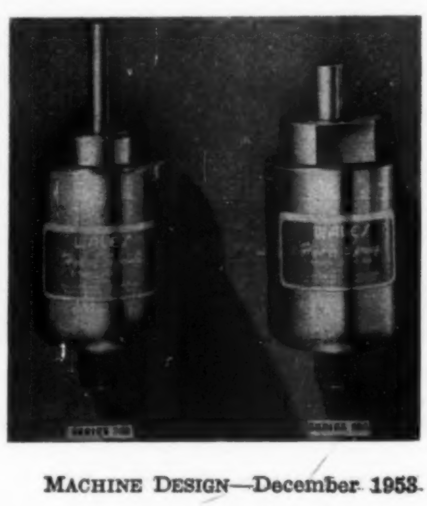
(*Continued from Page* 289)
switch and carries no current. All live metal parts are shielded. The unit can be mounted in any position and can be used to hold surface temperatures on heated plastic dies, melting pots, etc., or as a limit switch or alarm to control overheating. Setting to any temperature up to 550 F is accomplished by rotating a self-locking

adjustment screw which provides close adjustment. An adjustment shaft and knob can be provided when variable temperature control is desired. Operating parts are made of heat-resisting alloy metals, insulated with mica. Contacts are heavy-duty heat-dissipating electrolytic silver. Rated at 1500 w, 115-230 v ac, models HS and HK measure $2\frac{1}{2}$ x $\frac{3}{4}$ x $\frac{1}{2}$-in. high and $2\frac{1}{2}$ x $\frac{3}{4}$ x $\frac{5}{8}$-in. high, respectively. Made by **George Ulanet Co.,** 416 Market St., Newark 5, N. J.

For more data circle MD-110, Page 269

Hydraulic Springs

Up to 6000 and 8500 lb spring force can be provided in a relatively small space by Wales Hydra springs. They utilize the compressibility of fluids called Wales




Design Your Own
Precision Gaging Fixtures
. . . around this
Inexpensive
Gage Head
Cartridge
Measures by .0001" — .00001"
Compact, shock-proof, and moisture-proof, this Brown & Sharpe Gage Head Cartridge No. 953 gives you a simple means for designing high precision into your own gaging fixtures.
Readings in .0001" to .00001" are taken on the No. 950 Electronic Amplifier. Simply plug the cartridge into the amplifier. You take readings directly from the amplifier's accurately graduated dial . . . only one master required for setting. And you can design cartridges into several fixtures and use the same amplifier on all of them.
The unique design of the cartridge provides simple adjustment, frictionless movement, adjustable measuring pressure, and wide adaptability to many inspection devices, jigs, fixtures, and machines. Write for detailed Bulletin. Brown & Sharpe Mfg. Co., Providence 1, R. I., U. S. A.
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Comproils. Springs measure $2\frac{1}{4}$-in. diameter by $4\frac{1}{4}$-in. long and produce as much spring force as a 6-in. diameter by 8-in. long railroad car spring with $1\frac{1}{8}$-in. wire. Force and stroke can be adjusted by changing pre-loads or volumes or types of Comproils. Units are self-contained, preloaded and can be dampened and adjusted in place. Made by **Wales-Strippit Corp.**, Hydra Spring Div., 345 Payne Ave., North Tonawanda, N. Y.

For more data circle MD-111, Page 269

Steel Caster

Form-Forged caster has double ball bearing race for quiet, easy swiveling. Raceways are fully case-hardened, and the kingpin is extra heavy. Structurally shaped of heavy gage steel the caster withstands heavy loads in rough and

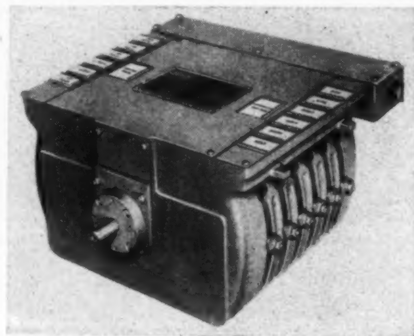
rugged service. Made in 5, 6, 8 and 10-in. sizes, with semi-steel, forged steel or rubber tread wheels, it has a load rating of up to 1500 lb and is available in either swivel or rigid types. Made by **Bassick Co.**, 437 Howard Ave., Bridgeport 2, Conn.

For more data circle MD-112, Page 269

Rotary Cam Limit Switch

Adjustable during operation, this switch features fine adjustment which is made by setting a thumbscrew on the outside of the housing. The large diameter swept by the cam faces offers added adjustment sensitivity. At the point of

adjustment of the thumbscrew, one degree is equivalent to about ⅛-in. of adjustment. Double-break snap switches permit accurate timing at the moment of contact. Make and break can be timed independently so that the circuit to be controlled can be closed from a fraction of a

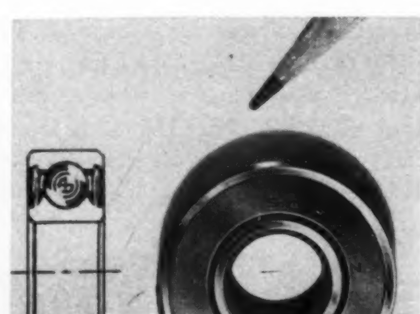


second to nearly a complete machine cycle. The switch operates in either direction of rotation. Two sizes are available, holding 12 cams, or up to 22 cams. Developed to meet the needs of mechanical press automation devices, the switch permits stroking of the press while individual switches are being adjusted. Made by **Danly Machine Specialties Inc.**, 2100 S. Laramie Ave., Chicago 50, Ill.

For more data circle MD-113, Page 269

Commercial Bearing

Commercial bearing, identified as CS 3258, for use in washing machines, farm equipment, small electric motors and other applications where speed and load are moderate,



has carburized and hardened raceways, ground surfaces and a molded plastic seal. Bearing is available in English or Metric sizes, with sizes ranging from ⅝ to 2.047-in. OD and bores from 3/16 to 1 in., with or without plastic seals and/or metallic shields. Up to 75 per cent of the load rating of a comparable precision bearing at 10,000 rpm is possible, depending upon materials specified. Made by **Schatz Mfg. Co.,** 6762 Fairview Ave., Poughkeepsie, N. Y.

For more data circle MD-114, Page 269

Magnetic Amplifier

Model M-22 stable push-pull magnetic amplifier has practically zero drift and an input impedance of 5, 25, 50 and 100 ohms. Signals of 0 to 20 millivolts dc produce a linear output from 0 to 10 milliamps dc into a load of 100 ohms. Operation is over temperature range from −50 to 70 C. The 60-cps model measures 2 x 2 x 2½ in., and 400-cps model is 1½ x 1½ x 2 in. Made by **Magnetic Controls Inc.,** 119 W. 63rd St., New York 23, N. Y.

For more data circle MD-115, Page 269

Heavy-Duty Engine

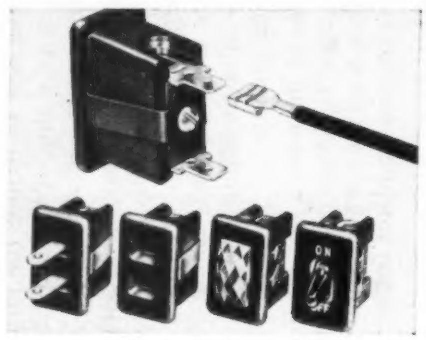
Delivering up to 26.8 hp at 3600 rpm, K-660 heavy duty gasoline engine is four-cycle, air-cooled, opposed-twin type. It weighs 225 lb, and standard equipment includes heavy duty muffler, diaphragm type

fuel pump, oil bath air cleaner and oil pressure gage. Air cooling system is designed to keep engine at correct operating temperature under all conditions. Governor is an externally mounted precision oil-bathed flyweight type, and is adjusted at 3000 rpm with load unless otherwise specified. Made by **Kohler Co.,** Kohler, Wis.

For more data circle MD-116, Page 269

Snap-In Electric Units

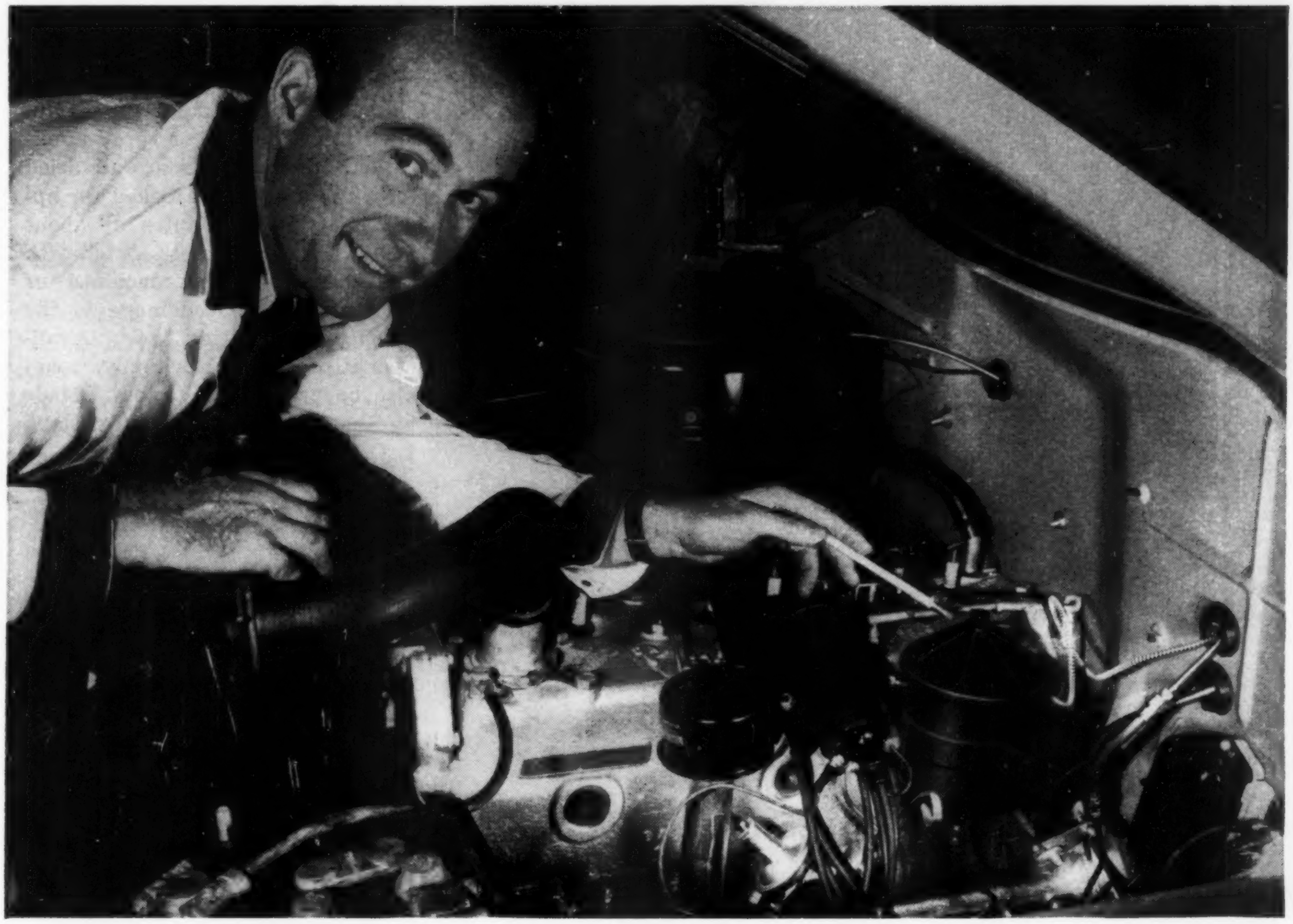
Diamond H Snap-In switches, outlets, pilot lights and interconnecting load plugs are now available with spade terminals for A-MP Quick Connect connectors as well as with standard screw terminals. Applicable to a wide range of electric equipment, appliances and machines, the devices are pushed into mounting holes where spring clips hold them firmly in place. Wiring is accomplished rapidly. Switches are rated 15 and 20 amp, 125 v; 10 amp, 250 v ac, and are also available with horsepower ratings. Pilots are rated 115 or 230 v ac. Made by **Hart Mfg. Co.,** 110 Bartholomew Ave., Hartford, Conn.



For more data circle MD-117, Page 269

Industrial Enamel

Emulsion type enamel employs water as both an ingredient and a reducing agent. It is nonflammable at normal room or operating temperatures, even if exposed to direct flame. The finish is not water soluble after application and is highly



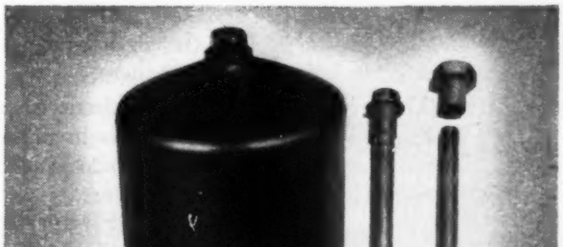
Here's where Purolator screened out rising costs

● Purolator makes oil filters for a prominent auto maker. They're top-notch filters that do a tough job well. Purolator and the car manufacturer are both proud of them.

Not long ago an RB&W "fastener engineer" got loose in the Purolator plant—just when company production executives were looking for a way to lick rising costs. He noticed that the Purolator filter was being assembled with a two-piece fastener made slowly and laboriously on a screw machine.

Our man told the Purolator people about RB&W's batteries of cold-forming machines. Purolator wanted to know more. Now their filter is assembled with a one-piece RB&W fastener that costs far less to make and assemble.

Chances are you can find a stage in your operations where RB&W "fastener engineering" can help you keep costs in line. As a leading manufacturer of all kinds of fasteners, we're always able to recommend and supply the right ones for all your needs. Write RUSSELL, BURDSALL & WARD BOLT AND NUT COMPANY, Port Chester, N. Y.



EASIER, FASTER ASSEMBLY undercut high costs when Purolator switched from a two-piece fastener *(right)* to an RB&W-designed cold-formed fastener *(left)* for its famous oil filter.

RB & W

108 YEARS MAKING STRONG THE THINGS THAT MAKE AMERICA STRONG

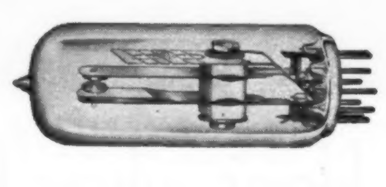
Plants at: PORT CHESTER, N. Y., CORAOPOLIS, PA., ROCK FALLS, ILL., LOS ANGELES, CALIF. *Additional sales offices at:* PHILADELPHIA, PITTSBURGH, DETROIT, CHICAGO, DALLAS, SAN FRANCISCO. *Sales agents at:* PORTLAND, SEATTLE. *Distributors from coast to coast.*

resistant to oil. Producing a high gloss finish, the enamel is formulated for either dip or spray application. Coated parts are air dried for 20 minutes, then baked for approximately 18 minutes at about 300 F. Industrial units coated with the enamel can be machined or fabricated without damage to the finish or the cutting tools. Available at present in black only, it will also be produced in colors. Made by **Sherwin-Williams Co.,** Cleveland 1, O.

For more data circle MD-118, Page 269

Delay Relays

Hermetically sealed miniature delay relays are not affected by altitude or humidity variation, withstand vibration, and are available with delays from 2 to 90 seconds. They can be supplied for



all standard heater voltages and have contact rating of 2 amp on 115-v noninductive ac. They are ambient compensated for temperatures from −50 to 70 C. Made by **Amperite Co. Inc.,** 561 Broadway, New York 12, N. Y.

For more data circle MD-119, Page 269

Bearing Alloy

Combining impact resistance and hardness, Frix bearing overlay material in electrode form is said to show wear resistance several times greater than that of conventional bronze welding rods or electrodes after a short heat treatment. It is a mixture of a soft mass with embedded crystals of wear resistant copper alloy. Deposits also possess elasticity and lubricating properties provided by lower melting point alloys included in the formulation. Film-like deposit can be heated to form an integrated bearing. Frix is made in range of

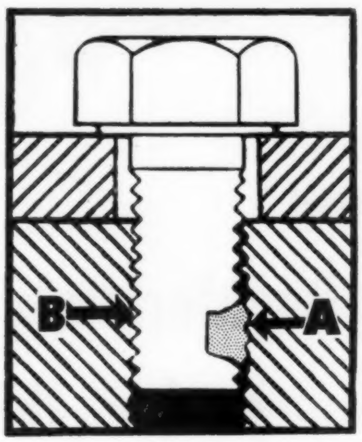
New Parts

sizes by **Eutectic Welding Alloys Corp.,** 40-40 172 St., Flushing, N. Y.

For more data circle MD-120, Page 269

Locking Screws and Bolts

Holtite-Nylok one-piece self-locking screws and bolts employ a nylon insert which conforms to the curvature of the screw threads to provide a locking action when stopped. Locking seated or unseated, the reusable screws and bolts act as a seal for gases or



liquids, do not harm mating thread of parts, resist vibration and can be used on such soft metals as aluminum and die castings without damaging them. They can be used at temperatures from −50 to 250 F. Photo shows nylon plug (A) which sets up lateral thrust to smoothly wedge mating threads together (B). Made by **Continental Screw Co.,** New Bedford, Mass.

For more data circle MD-121, Page 269

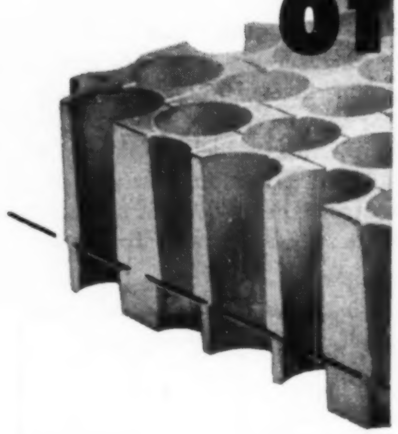
Plastic-Metal Laminate

Made by rolling rigid or semi-rigid Marvinol vinyl sheet onto heated metal to which a special adhesive coating has been applied, Marvinol-Metal laminate has higher abrasion resistance than varnish, paint and baked enamel finishes. The laminate can be sheared, drilled or punched without chipping. Crimp rolls, 90-degree crimp bends, 180-degree bends and deep draws can

(*Continued on Page* 308)

Here's a honey

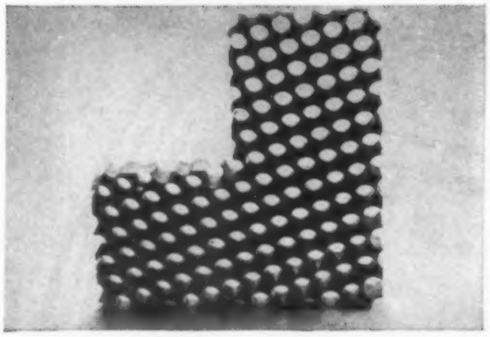
of



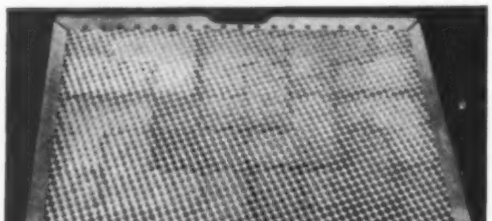
and here's how

Doehler-Jarvis

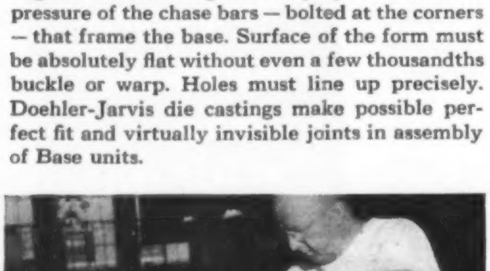
perfect this



This is the "L"-shaped die cast unit just as it comes from Doehler-Jarvis machines. It is 5" x 5" on the long sides, 2½" on the four short sides and is ¾" thick. Prior to Base assembly at the Blatchford plant, flash is removed from the edges, and top and bottom are shaved slightly. No other finishing is needed . . . thanks to Doehler-Jarvis die-casting methods.



Here is a complete Blatchford "Honeycomb" Base with inside chase dimensions of 15" x 20". (Unit assembly of these Bases makes possible sizes up to 50" x 75".) It is made up of 16 of the angle blocks held together only by the enormous pressure of the chase bars — bolted at the corners — that frame the base. Surface of the form must be absolutely flat without even a few thousandths buckle or warp. Holes must line up precisely. Doehler-Jarvis die castings make possible perfect fit and virtually invisible joints in assembly of Base units.



Stonehands position and anchor printing plates on the "Honeycomb" Base with special catches that fit and expand in the holes. The thin walls of the Doehler-Jarvis castings have to be strong and solid to withstand both the locking pressure and the gripping forces exerted by the catch. When the plate mounting is completed, the entire "locked-up" form goes to the printing press.

Before this intricate section was perfected, E. W. Blatchford Co. had to overcome some real production headaches.

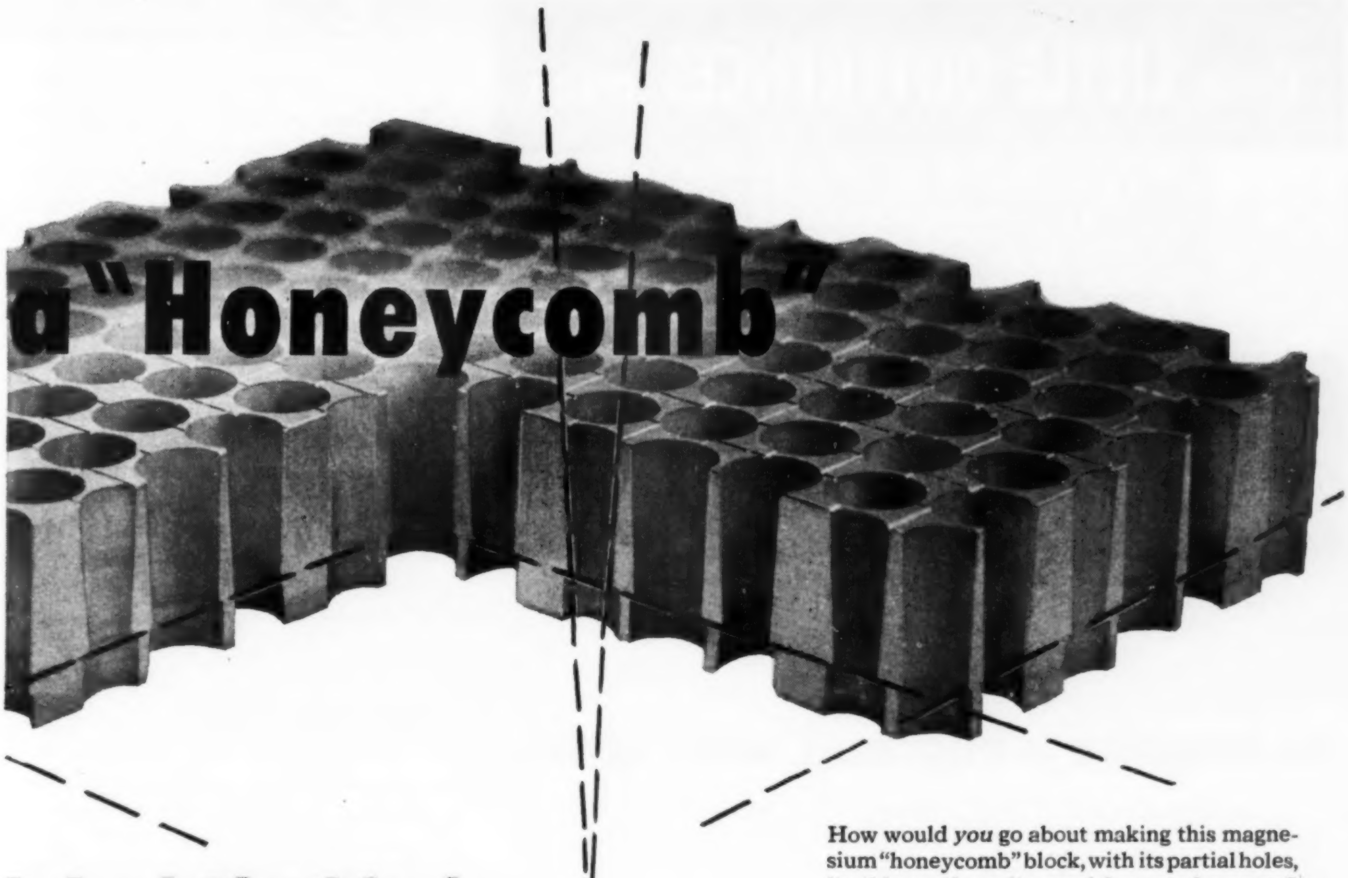
To machine their lightweight "Honeycomb" plate mounting system for printers would have been prohibitively costly. They would have had to remove 71% of the metal volume in the component blocks. And no one could agree on a way to form the partial holes along the "block" edges without running up rejects and tool breakage.

Then someone suggested die casting... a production technique that was just beginning to get wide attention 35 years ago. Die casting proved to be the practical answer but Blatchford engineers were still not completely satisfied. The behavior of the alloy was not always predictable and warpage of the "blocks" occasionally caused trouble.

Then Doehler-Jarvis entered the picture and worked the problem over with Blatchford people.

That did it. Quality went up and product performance improved. Over the years Doehler-Jarvis cast untold thousands of the "honeycomb" units. Printers bought Blatchford "Patent Base" like hotcakes... under Blatchford's 10-year guarantee.

Then, just a few years ago, Blatchford wanted to make the Base even lighter on men, even lighter on power, even lighter on presses. Again engineers from both companies joined forces. They put several light metal alloys to the toughest of tests.



How would *you* go about making this magnesium "honeycomb" block, with its partial holes, its thin sections, its precision requirements?

helped Blatchford complex component

A magnesium alloy finally was selected and Doehler-Jarvis designed and built a new production die.

Today, so precise is the Doehler-Jarvis casting technique, only 1/10 of an ounce – 1.82% of the as-cast weight – is removed in finishing the "Honeycomb" block. The new Magnesium Base weighs 72% less than the original Zinc Base. Needless to say, it too is winning its battle in the market place.

As one Blatchford executive puts it, "Doehler-Jarvis die castings helped get our Base over the hump in the first place, helped us stay competitive over the years, then helped us give the product a shot in the arm."

In its half-century of die-casting, Doehler-Jarvis has helped turn good ideas into metal for many producers, large and small – among them some of the best known makers of automotive, electrical, communication, household, office and other types of equipment.

When they come to Doehler-Jarvis, they know that Doehler-Jarvis can supply both the resources and the resourcefulness they need.

So next time you have a good new product idea... call Doehler-Jarvis in on the production engineering. We may be able to give you just the help you need.

Doehler-Jarvis Division
of
National Lead Company
General Offices: Toledo 1, Ohio

*Reg. U. S. Pat. Off.

New Parts

(*Continued from Page* 305)

be made without damaging the coating. Tests indicate adhesion in excess of 40 lb per sq in. of width. Cold-rolled steel, hot-rolled pickled steel and aluminum, from 18 to 34 gage, have been used; thickness of vinyl used ranges from 0.002 to 0.02-in. Available in colors, the vinyl can be applied to metal for use in machine housings; corrosion resistant chemical containers, piping and ducts; truck and trailer body panels and furniture. Developed by Naugatuck Chemical Div., **United States Rubber Co.**, Rockefeller Center, New York 20, N. Y.

For more data circle MD-122, Page 269

Time Delay Relays

Applicable to control circuits requiring two time delay steps or momentary impulses, NET Agastat relays incorporate an adjustable auxiliary switch. Overall time delay is adjustable from 0.1 second to 10 minutes or more, and auxiliary switch can be set to operate at any time from 0 to 15 seconds after the start of the overall period. Four models are available: No. 11 and 12 are single-pole, double-throw, double-break, and No. 21 and 22 double-pole, double-throw single-break types. Relay switch rating is 15 amp at 115 v and auxiliary switch rating is 10 amp at 115 v. Many different op-

erations, from momentary impulses to complex delay-contact sequences, are possible. Made by A'G'A Div., **Elastic Stop Nut Corp. of America,** 1027 Newark Ave., Elizabeth, N. J.

For more data circle MD-123, Page 269

Tanks

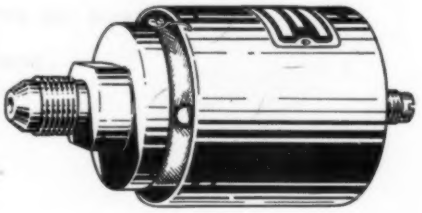
The tanks illustrated are a few of many styles that can be made to special requirements of 14 to 18-gage steel in sizes ranging from 1 to 150 gallons. Features frequently included are baffles, settling and collecting trays, baskets, filter

screens, hinged covers, shelves, covers with safety devices, casters, etc. Standard designs also are available. Typical applications include coolant and filtering systems, degreasing, soaking and dipping, oiling systems, hydraulic systems, and others. Made by **Graymills Corp.,** Dept. C, 3705 North Lincoln Ave., Chicago 13, Ill.

For more data circle MD-124, Page 269

Relief Valve

Metallic diaphragm of Model RD-10280 isobaric relief valve balances out the effect of atmospheric pressure so that the valve opens at exact preset system pressure under conditions of varying altitude. The diaphragm withstands temperature from −65 to 165 F. With only one connection to the

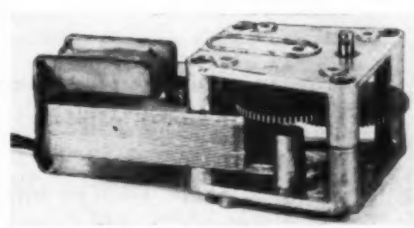


air system, the valve is easily adjusted for blow-off pressure. Port for tube connection is 7/16-20NF-3 thread per AND10056, style E. The exhaust opens to ambient air. Cracking and reseating of the valve are accomplished within 2 psi of the valve setting. The cracking pressure may be set very close to the maintained system pressure without loss of air through the relief valve. Pressure range is adjustable from 25 to 40 psi absolute. The valve weighs ¼-lb. Made by Lear-Romec Div., **Lear Inc.,** Elyria, O.

For more data circle MD-125, Page 269

Induction Motor

Model SP-6 shaded pole induction motor features a compact gear train. Geared to 6 rpm with a torque of 20 in.-lb, the motor operates on 110 v ac, 50/60 cycles only. Other models are available



from 1 to 1155 rpm. The compact motor measures 4¾ x 2½ x 2 in. overall. Uses include actuators, electrical appliances, dispensers, vending machines. Made by **General Die & Stamping Co.,** 262 Mott St., New York 12, N. Y.

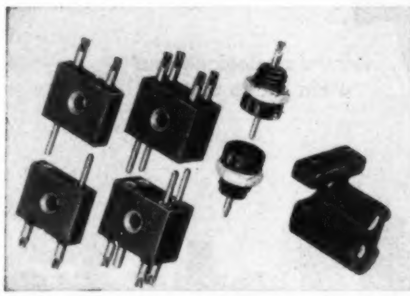
For more data circle MD-126, Page 269

Miniature Connectors

Countersunk side mounting arrangement makes special miniature rectangular two and four-contact reverse pin and socket connectors adaptable to small equipment. The one-pin round connector makes a compact and simple one-wire disconnect, can also be used as a disconnect type feed-through or standoff, or can be converted into a small, light and simple coaxial con-

New Parts

nector. Four special connectors are available with mineral-filled Melamine; Plascon glass fiber reinforced alkyd, type 440-A; or diallyl phthalate insulation. Contacts are silver and gold plated and

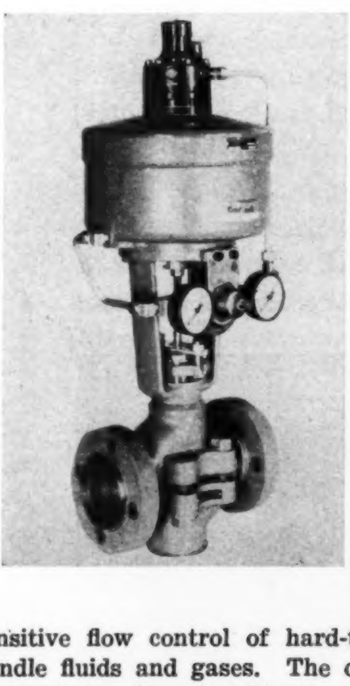


normally have solder wells for No. 20 AWG wire. They are also available in standard pin and socket arrangements from **DeJur-Amsco Corp.**, 45-01 Northern Blvd., Long Island City 1, N. Y.

For more data circle MD-127, Page 269

Control Valve

Actuated by a piston-type air motor, LB series control valves are designed to fulfill exacting requirements for accurate, positive and



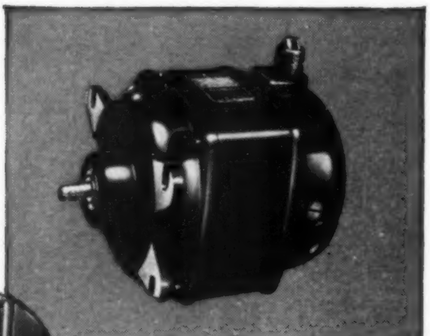
sensitive flow control of hard-to-handle fluids and gases. The operator is capable of positioning re-

(*Continued on Page* 318)

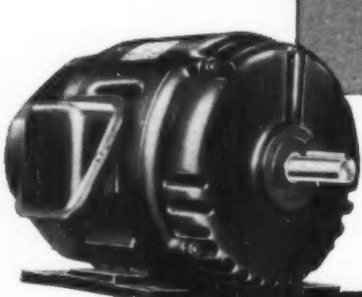
When the specs call for...

Explosion-Proof Motors.

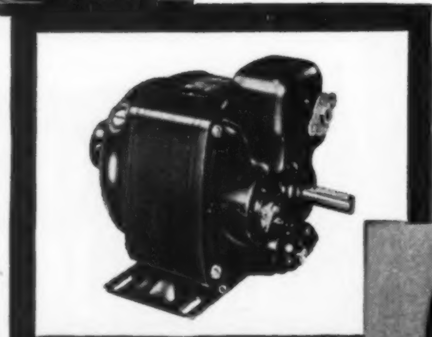
Leland explosion-proof ether pump motor.



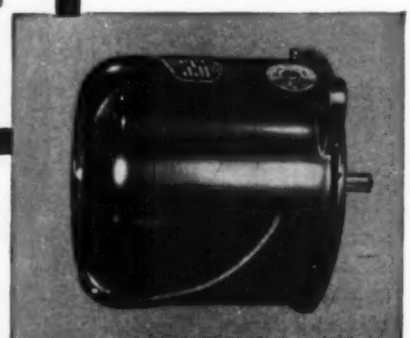
Leland explosion-proof hose reel motor.



Leland repulsion-start, induction-run standard motor in drip-proof open frame provides high starting torque for many general-purpose applications.



Leland explosion-proof gasoline pump motor.



Leland flange-mounted oil burner motor.

See AMF's new, exciting television show, "OMNIBUS", every Sunday. Check local paper for time and channel.

New Parts

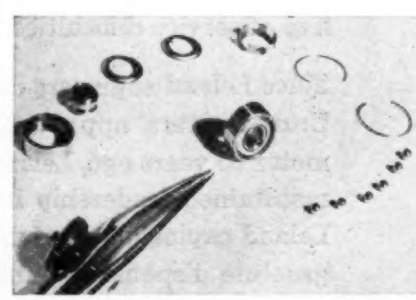
(Continued from Page 315)

versible venturi type Conoplug accurately to well over one part in 1000 of instrument output span. Sizes up to and including 4 in. are available for series 15, 30, and 60 body ratings. Both straight through and angle valve bodies are standard in carbon steel, bronze, aluminum, stainless steel, Hastelloy and Durimet. Valves are suitable for services including steam and the most corrosive chemical conditions at up 2500 psi, 1000 F and high vacuums. Made by **Conoflow Corp.**, 2100 Arch St., Philadelphia 3, Pa.

For more data circle MD-128, Page 269

Shielded Miniature Bearings

Series of single and double-shielded miniature precision ball bearings of both full race and retainer type construction as small as 5/32-in. OD are offered in nine sizes. Shields, which are removable, are recessed to avoid interference with shaft shoulders or housings, allowing full freedom in tolerances. Shields on the outer

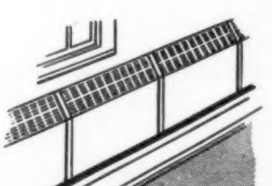


circumference fit into a recess in the outer race of the bearing and are positively positioned by a retaining snap ring. The inner circumference of the shield fits close to the shoulder of a relieved land on the inner race, permitting a clearance of less than 0.002-in. Produced in both chrome bearing steel and stainless, with all shields of stainless steel, the bearings may be used in precision mechanisms such as miniature blowers and small electric motors and wherever lint, dust or other foreign matter

FASTEST THING IN FASTENINGS®

Tinnerman
SPEED GRIP®
Nut Retainers...
snap into place by hand. Provide heavy-duty fastenings . . . eliminating welding, clinching, or staking. Reduce materials and materials handling; ideal for blind locations. Complete range of sizes available!

radiant·ray saves 78% with Tinnerman SPEED GRIPS®



Reducing the cost of assembling enclosures to Airfoil Finned heaters was a double problem for Engineers at Radiant-Ray Radiation, Inc., New Britain, Conn. Easier ways of installing the units had to be considered also. On both counts SPEED GRIP nut retainers turned the trick over all other methods! These SPEED NUT brand fasteners saved 78% in production time, machining costs, and materials . . . permitting the use of less costly channel, eliminating punching and tapping of holes on wall hangers, and doing away with the longer flange on the expansion cradle hanger.

One leading contractor has this to say about easier, faster custom-fitting and installing: "Adjusting and levelling are now a 'snap' for us due to these SPEED GRIPS." Your Tinnerman representative is trained to help you spot savings opportunities like this on your product assemblies. See him soon!

Write today for your "SPEED NUT Savings Stories" booklet of typical Tinnerman savings to industry: TINNERMAN PRODUCTS, INC., BOX 6688, Dept. 12, Cleveland 1, Ohio.

In Canada: Dominion Fasteners, Ltd., Hamilton, Ont. *In Great Britain:* Simmonds Aerocessories, Ltd., Treforest, Wales. *In France:* Aerocessoires Simmonds, S. A.—7 rue Henri Barbusse, Levallois (Seine).

are present. Made by **Miniature Precision Bearings Inc.,** Keene, N. H.

For more data circle MD-129, Page 269

Miniature Motor

Measuring $1\frac{5}{8}$ in. in diameter, $2\frac{3}{8}$ in. in length and weighing approximately 8 oz, this miniature motor is rated at 0.002-hp at a speed of 6500 rpm. Extremely sensitive, it can be controlled by the output of two small vacuum tubes. Of the split-shunt, reversible rotation type, drawing maximum armature current of 0.8-amp

from a 28-v line, the unit is capable of responding to field currents of 0.00075-amp. It operates at altitudes of 50,000 ft and at temperatures as low as −65 F. It is unaffected by dry, hot, or humid environmental conditions. Made by **General Electric Co.,** Schenectady 5, N. Y.

For more data circle MD-130, Page 269

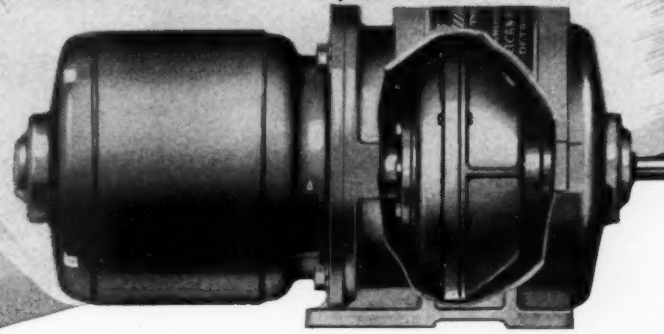
Silicone Rubber

Silastic 50 and Silastic 80 are high tensile, high elongation silicone rubber materials which have good compression set characteristics, are highly resistant to water absorption and retain their properties at temperatures ranging from −80 to 500 F. When cured, their respective Shore A hardnesses are 50 and 80. White, semitranslucent materials that can be colored to practically any intensity by the use of special coloring pigments, each is a completely formulated stock that can be used alone to fabricate finished parts. They can

"POWER IN A PACKAGE"

GÝROL FLUID DRIVE USES PARKER DIE CASTINGS

to Reduce COST, *Add Strength, Minimize Machining*



AMERICAN BLOWER'S Gýrol Fluid Drive Power Unit is another product using Parker Die Castings to achieve greater strength while reducing cost. The two rotating elements with die cast blades shown at the left transmit the power through a vortex of oil. Alloy aluminum is used for light weight and high strength. Machining is minimized because the blades are die cast accurately and cleanly. Gýrol Fluid Drive units are used profitably in giant utilities, steel mills, paper plants, automotive and many processing plants. These units efficiently power many industrial machines—conveyors, centrifugals, cranes, mixers, winders, spinners and others where smooth acceleration, overload protection and shock absorption are desirable. These die castings by Parker, adding strength, cutting weight and minimizing machining costs, are vital elements in the top performance of Gýrol Fluid Drive units. Call Parker on your next die casting requirements. Our engineers will work with you to our mutual benefit.

and when you think of Die Castings THINK OF

Parker White-Metal Company • 2153 McKinley Ave. Erie, Penna.

PARKER ALUMINUM *and* ZINC *Die Castings*

also be blended with each other to yield material having any desired hardness between 50 and 80, the hardnesses being proportional to the amount of each stock used. Suitable for fabrication into a variety of parts, the materials can be molded, extruded, laminated and calendered. Typical uses include molded parts, tubing, gaskets, belts, electrical insulating parts, and wire and cable. Developed by **Dow Corning Corp.**, Midland, Mich.

For more data circle MD-131, Page 269

Solenoid

Handling wide temperature variations, this "hot" solenoid supplies positive, actuating force at ambient temperatures from −65 to 500 F. It features a high-temperature

switch, special heat resistant wiring and insulation. Future models are expected to have an effective operating range beyond 600 F. Manufactured by **PSP Engineering Co.**, 8420 Otis St., South Gate, Calif.

For more data circle MD-132, Page 269

Stainless Steel Coating

Applied by brushing or spraying, Proco liquid stainless steel coating

New Parts

offers the corrosion and moisture resistance of 18/8 No. 302 type stainless steel. It covers plywood, plaster, linoleum, metal work and other surfaces with the blue-gray finish of stainless steel. Surfaces coated with the material have a tough, continuous, corrosion resistant finish. A gallon of the coating covers 300 to 400 sq ft. Available from **Protective Coatings Inc.,** 807 North Fremont Ave., Tampa 6, Fla.

For more data circle MD-133, Page 269

Subminiature O-Rings

Designed to meet precise space requirements, series 6227 close tolerance, subminiature elastomeric O-rings are available for use in pumps, valves, nozzles and in general hydraulic applications. Stock sizes are 0.029 x 0.040 x 0.109-in., 0.070 x 0.040 x 0.150-in. and 0.1 x 0.07 x 0.24-in. for moving or nonmoving seals. Rings withstand wide range of pressure, temperature and movement. They seal in both directions of pressure flow, are durable and have low running friction. Made by **Stillman Rubber Co.,** Dept. D., 5811 Marilyn Ave., Culver City, Calif.

For more data circle MD-134, Page 269

Phosphor-Bronze Strip

Precision rolled to very close tolerances and to thin gages and foils, phosphor bronze strip possesses high tensile and yield strength, good ductility, resiliency, high fatigue strength, wear resistance and good bearing qualities. It is custom rolled in strip from 0.0005 to 6 in. wide, to tolerances of $\pm$0.0001-in. Typical uses include bellows and diaphragms in aircraft, heating, air conditioning and refrigeration components; high strength springs, snap switches, socket and plug contacts; printed electronic and communication circuits; lock washers, sleeve bushings, and clutch disks. Made by **American Silver Co. Inc.,** Flushing, N. Y.

For more data circle MD-135, Page 269

PUMPS *and* MOTORS

More Power For Your Money!

- Fixed displacement pumps and motors . . . plain, foot, flange and motor mountings . . . capacities 1 to 120 G.P.M. . . . 1000 P.S.I.
- Variable displacement pumps and motors . . . radial piston type . . . capacities 1 to 185 G.P.M. . . . 2500 to 3000 P.S.I.
- Ample selection of pump or motor controls.

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ACCESSORIES POWER UNITS VALVES CYLINDERS

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THE HYDRAULIC PRESS MFG. CO.
MOUNT GILEAD, OHIO, U. S. A.
1096 MARION ROAD

A Factor in the Field of Hydraulics Since 1877!

ONE SOURCE FOR ALL COMPONENTS!

ENGINEERING DEPARTMENT EQUIPMENT

Data Recorder

This recorder takes 100 identifiable readings on an 8½ x 11-in. chart which can be removed and replaced with accurate indexing, permitting the collection, on a single chart, of test data taken at different times. An unskilled operator can take up to 600 bridge

readings an hour. The instrument can be used for any bridge or potentiometric measurement. The print drive servo has a sensitivity of 25 mu v; full scale spans as low as 15 mv are practical. Automatic test cycle requires one minute. Made by **Barnes Development Co.,** 213 West Baltimore Pike, Lansdowne, Pa.

For more data circle MD-136, Page 269

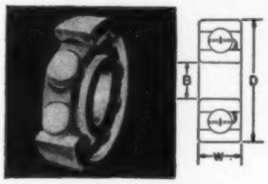
Analog Computing Unit

Embodying a basic electronic-analog facility, model HK operational manifold component is comprised of a set of 10 operational amplifiers plugged into a special chassis. All common service connections to the amplifier, such as those for supply voltages, are provided within the unit. Zeroing means of improved stability are accessible at the front. The unit has a symbolic arrangement of computing connections whereby

for YOUR EVERY USE...

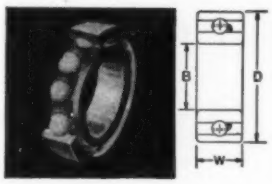
SPECIFY *Micro* BEARINGS

RADIAL TYPES



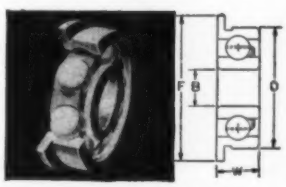
Retainer

BEARING	B	D	W
R0	3/64-.0469	5/32-.1562	1/16-.0625
R1	.0550	3/16-.1875	5/64-.0781
R1-4	5/64-.0781	1/4-.2500	3/32-.0937
R1-5	3/32-.0937	5/16-.3125	7/64-.1094
R2-5	1/8-.1250	5/16-.3125	7/64-.1094
R2	1/8-.1250	3/8-.3750	5/32-.1562



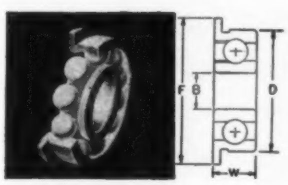
Retainer Extra Light

R133	3/32-.0937	3/16-.1875	1/16-.0625
R144	1/8-.1250	1/4-.2500	3/32-.0937
R155	5/32-.1562	5/16-.3125	7/64-.1094
R156	3/16-.1875	5/16-.3125	7/64-.1094
R166	3/16-.1875	3/8-.3750	1/8-.1250
R168	1/4-.2500	3/8-.3750	1/8-.1250
R188	1/4-.2500	1/2-.5000	1/8-.1250
R613M	6mm.	13mm.	3.5mm.



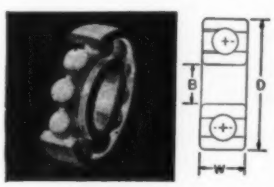
Flanged Retainer

FR1	.0550	3/16-.1875	5/64-.0781
FR1-4	5/64-.0781	1/4-.2500	3/32-.0937
FR1-5	3/32-.0937	5/16-.3125	7/64-.1094
FR2-5	1/8-.1250	5/16-.3125	7/64-.1094
FR155	5/32-.1562	5/16-.3125	7/64-.1094
FR156	3/16-.1875	5/16-.3125	7/64-.1094
FR166	3/16-.1875	3/8-.3750	1/8-.1250
FR188	1/4-.2500	1/2-.5000	1/8-.1250



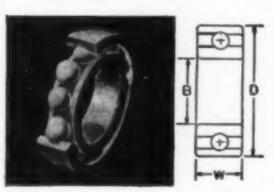
Flanged Full

F13	.0550	3/16-.1875	5/64-.0781
F14	5/64-.0781	1/4-.2500	3/32-.0937
F15	3/32-.0937	5/16-.3125	7/64-.1094
F154	1/8-.1250	5/16-.3125	7/64-.1094
F155	5/32-.1562	5/16-.3125	7/64-.1094
F156	3/16-.1875	5/16-.3125	7/64-.1094



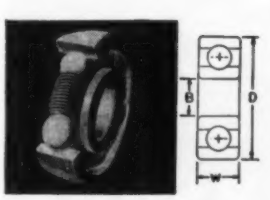
Full

10	.0250	.1000	1/32-.0312
12	.0400	1/8-.1250	3/64-.0469
12½	3/64-.0469	5/32-.1562	1/16-.0625
13	.0550	3/16-.1875	5/64-.0781
14	5/64-.0781	1/4-.2500	3/32-.0937
15	3/32-.0937	5/16-.3125	7/64-.1094
16	1/8-.1250	3/8-.3750	5/32-.1562



Full Extra Light

144	1/8-.1250	1/4-.2500	3/32-.0937
154	1/8-.1250	5/16-.3125	7/64-.1094
155	5/32-.1562	5/16-.3125	7/64-.1094
156	3/16-.1875	5/16-.3125	7/64-.1094
157	7/32-.2187	5/16-.3125	7/64-.1094
168	1/4-.2500	3/8-.3750	1/8-.1250
1810	5/16-.3125	1/2-.5000	5/32-.1563



Spring Retainer

R1-4Z	5/64-.0781	1/4-.2500	3/32-.0937
R1-5Z	3/32-.0937	5/16-.3125	7/64-.1094
R2-5Z	1/8-.1250	5/16-.3125	7/64-.1094
R2Z	1/8-.1250	3/8-.3750	5/32-.1562
R155Z	5/32-.1562	5/16-.3125	7/64-.1094
R156Z	3/16-.1875	5/16-.3125	7/64-.1094
R166Z	3/16-.1875	3/8-.3750	1/8-.1250
R168Z	1/4-.2500	3/8-.3750	1/8-.1250

MICRO makes, and makes possible

• • • **NEW PRODUCTS**

► By SIZE, TYPE, and MATERIAL • **MICRO** provides the widest selection of small ball bearings made to precision quality.

► DIMENSIONS of complete assemblies range from ½" overall diameter down to .1000".

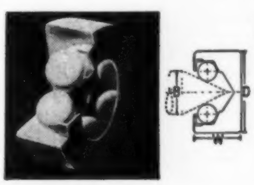
Seven RADIAL and three other series are standard in chrome and stainless steels with some available in beryllium copper. SPECIAL SIZES and MATERIALS are also considered.

► TOLERANCES are ABEC5 or BETTER.

NEW • • •

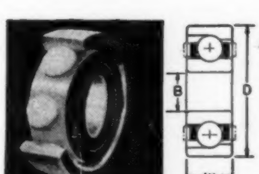
► The **MICRO** RETAINER (R) bearings, first developed by NHBB in the U. S., are a significant improvement for sizes previously available only in the full type. The retainer produces smoother running and lower friction. The crown retainer has long been in use, and many sizes are also being equipped with the new ribbon type — a two-piece cylindrical-pocket, cone-controlled and balanced retainer. ►

OTHER TYPES



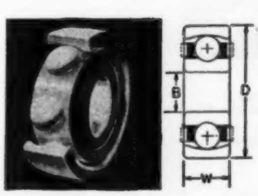
Pivot

BEARING	B	D	W
22½M	.040	4mm.	2.4mm.
23M	.050	5mm.	3.2mm.
24	.085	1/4-.2500	1/8-.1250
24H	.085	1/4-.2500	1/8-.1250
26	.165	3/8-.3750	3/16-.1875
26H	.165	3/8-.3750	3/16-.1875



Angular Contact

113	.0550	3/16-.1875	5/64-.0781
114	5/64-.0781	1/4-.2500	3/32-.0937
115	3/32-.0937	5/16-.3125	7/64-.1094
1154	1/8-.1250	5/16-.3125	7/64-.1094
116	1/8-.1250	3/8-.3750	5/32-.1562



Self Aligning

214	5/64-.0781	1/4-.2500	3/32-.0937
215	3/32-.0937	5/16-.3125	7/64-.1094
2154	1/8-.1250	5/16-.3125	7/64-.1094
216	1/8-.1250	3/8-.3750	5/32-.1562

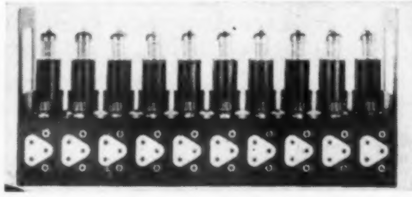
NEW HAMPSHIRE SPECIFY *Micro* BEARINGS **BALL BEARINGS, INC.**

41 MICRO CIRCLE • PETERBOROUGH, N. H. • TELEPHONE 424

Write for Catalog 53 for complete data on the more than 130 available **MICRO** bearings, or consult our Design Engineering Department.

Engineering Equipment

circuit elements and input and output leads are readily plugged in to form feedback circuitry. Of the four amplifier poles comprising plus input, minus input, output and ground, the jacks for every pair are spaced on ¾-in. centers for double banana plugs. Speed

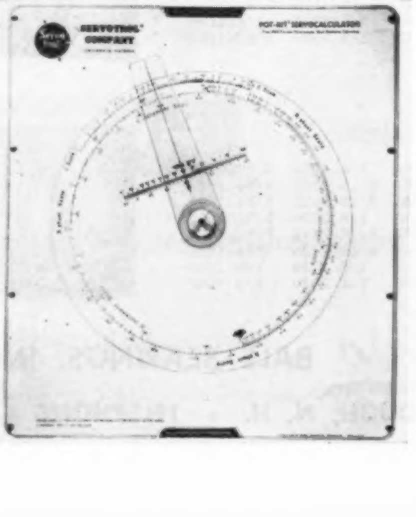


of computation ranges from a repetitive or single-shot period of 0.001-second to a solution time of a number of minutes. Power requirements are plus and minus 300 v dc at 50 ma and 115 v ac at 0.5 amp. Positive and negative inputs to each amplifier, both at high impedance, are available. Made by **George A. Philbrick Researches Inc.**, 230 Congress St., Boston 10, Mass.

For more data circle MD-137, Page 269

Disk Calculator

This device computes the combined potentiometer resistance and shunt resistor values required in converting a linear potentiometer to a nonlinear function. Values corresponding to maximum increment of resistance change are located on the C scale. The four increments of resistance change be-



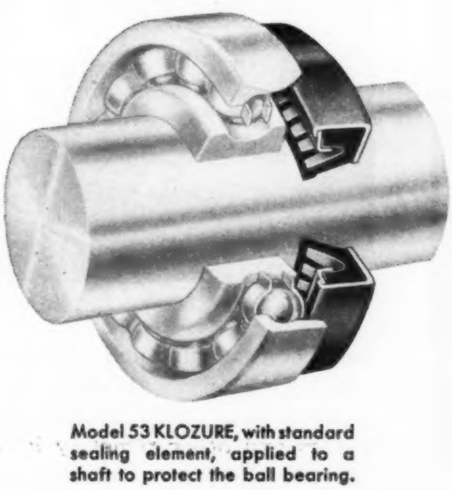
Dependable Bearing Protection for *All* Service Conditions

The true value of any oil seal must be measured by the ability of the sealing element to protect the bearing by keeping the lubricant *in*, dirt and moisture *out*.

For your bearing protection Garlock KLOZURES offer you these important sealing-element advantages:

1. **Synthetic rubber sealing elements** for oil, grease, water, mild acids and alkalies at temperatures up to 300° F.
2. **Silicone rubber sealing elements** for extremes of high and low temperatures.
3. **Teflon sealing elements** for strong acids.
4. **Choice of sealing element designs** with finger spring or garter spring for light, medium or heavy duty service at all speeds.
5. **All sealing elements** accurate and uniform in size, non-porous, tough, durable, non-abrasive and free-running.

Garlock KLOZURE Oil Seals are made in a complete range of sizes and in many models. For complete information call your Garlock representative or write for KLOZURE Catalog No. 10.



Model 53 KLOZURE, with standard sealing element, applied to a shaft to protect the ball bearing.

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THE GARLOCK PACKING COMPANY, PALMYRA, NEW YORK
In Canada: The Garlock Packing Company of Canada Ltd., Toronto, Ont.
Branch Offices in Most Principal Cities

GARLOCK

PACKINGS, GASKETS, OIL SEALS,
MECHANICAL SEALS,
RUBBER EXPANSION JOINTS

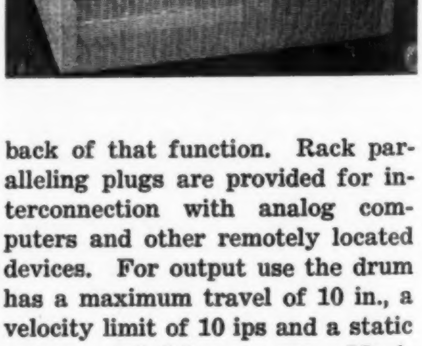
Engineering Equipment

tween taps which have been determined on a graph are located on the rotor which is locked or aligned with a hairline, permitting a reading of the value of the R shunt scale opposite the arrow. This provides the value of the shunt resistor to the total desired resistance. A pull-out slide provides instructions for use as well as a series of typical applications, such as one or two end resistors, slope reversals at zero or maximum resistance, slope reversals not at zero or maximum resistance, or double slope reversal. Available from **Servotrol Co.,** Framingham Centre, Mass.

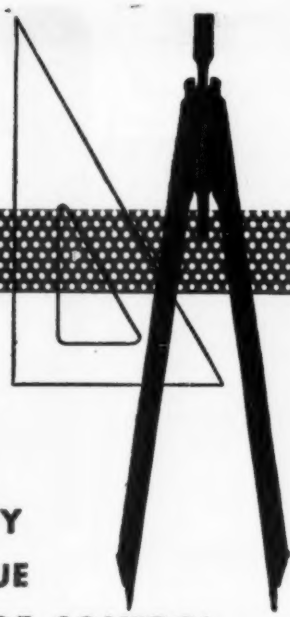
For more data circle MD-138, Page 269

Input-Output Instrument

Model T1 GEDA input-output device records or plays back one variable voltage as a function of a second variable voltage. Recording of an output function may be followed immediately by a play-



back of that function. Rack paralleling plugs are provided for interconnection with analog computers and other remotely located devices. For output use the drum has a maximum travel of 10 in., a velocity limit of 10 ips and a static accuracy of 0.2 per cent. Maximum pen travel is 7½ in.; velocity limit is 7 ips; and static accuracy is 0.2 per cent. Frequency response of both drum and pen represents a phase lag of less than 5 degrees at one cycle per second with a peak-to-peak amplitude of 2 in. Scale factors are 10, 5, 1, 0.5 and



for
- SAFETY
- FLEXIBILITY
- REVERSIBILITY
- HIGH TORQUE
- SIMPLICITY OF CONTROL

design with

GARDNER-DENVER

MULTI-USE AIR MOTORS

Fully described in Bulletin AM-1.
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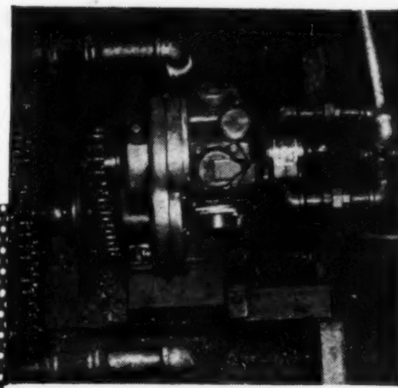
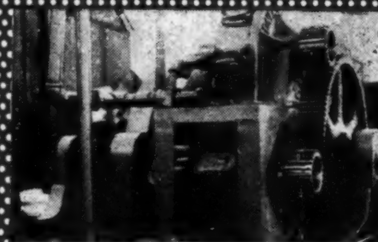
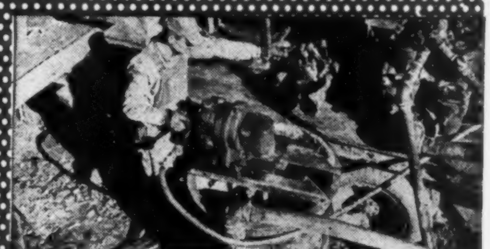


(5) steering tugboat

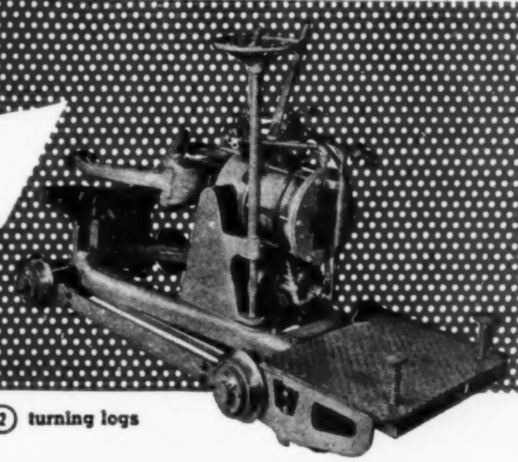


(4) hoisting ore

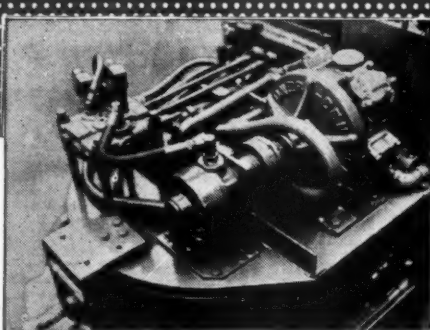


(3) mixing grout

MULTI-USE AIR MOTORS PUT TO WORK...



(2) turning logs



(1) driving hydraulic pump

SINCE 1859

GARDNER-DENVER

Gardner-Denver Company, Quincy, Illinois
In Canada: Gardner-Denver Company (Canada), Ltd.,
14 Curity Avenue, Toronto 13, Ontario

THE QUALITY LEADER IN COMPRESSORS, PUMPS AND ROCK DRILLS FOR CONSTRUCTION, MINING, PETROLEUM AND GENERAL INDUSTRY

0.1 v per in. and 50, 10, 5, 1 and 0.5 mv per in. As an input device the unit has an output voltage of 0 to 75 and a scale factor of 10 v per in. Static accuracy is 0.2 per cent. Output voltage is correct to within 0.7-v if center of the pickup element is not forced to lag the curve by more than 0.5-in. Power requirements are 105 to 125 v, 60 cycles and 200 w. Made by **Goodyear Aircraft Corp.**, Akron, O.

For more data circle MD-139, Page 269

Rolling Drawing Instrument

Transparent Glide-Rule combines triangles, T-squares, straight edge, scales, protractor and parallel rules in a 2½ x 9-in. instrument. It has accurate, engraved scales and protractor and four drawing edges

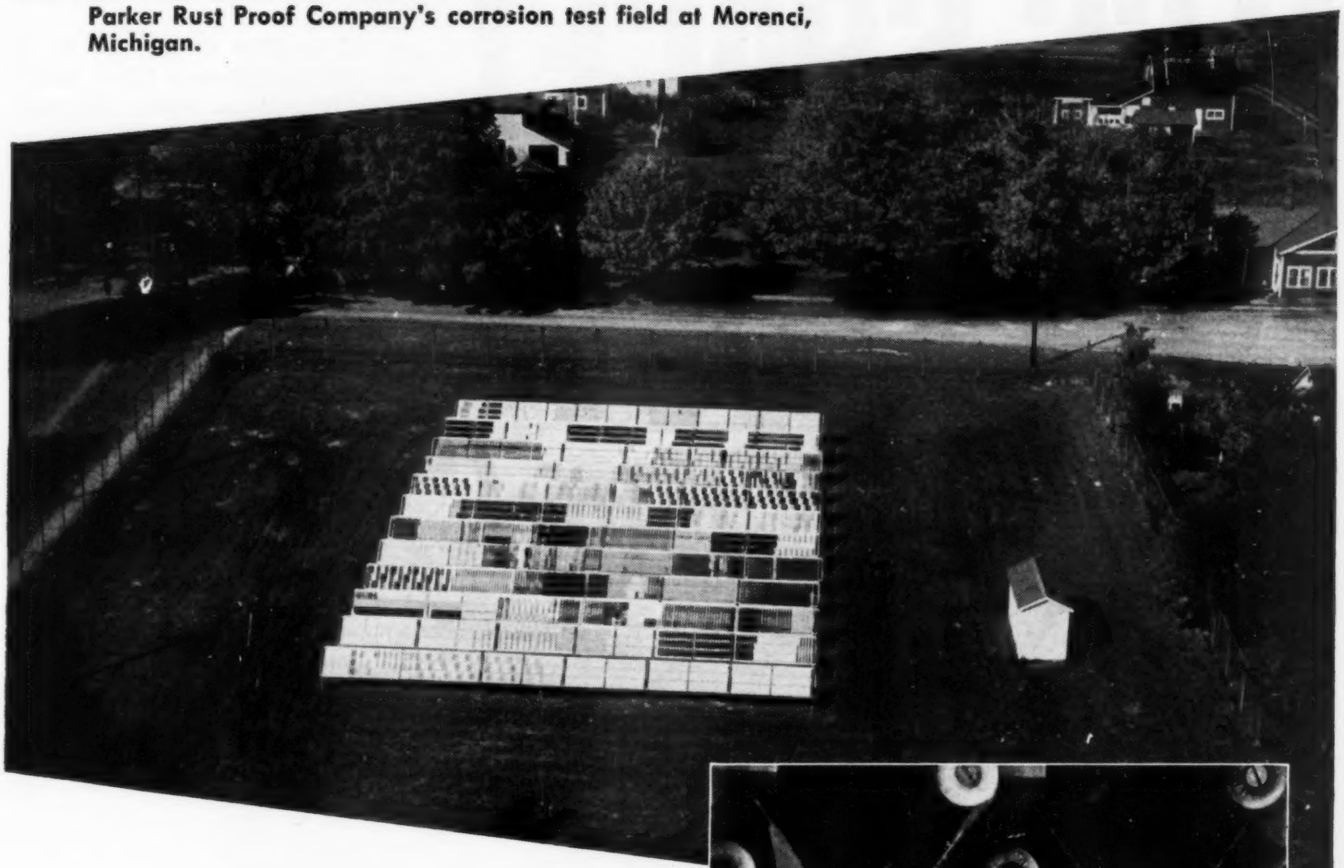
with scales graduated in 1/16 and 1/20-in. The protractor is a full 180 degrees, with emphasized 30, 45 and 60-degree angles. Capable of rolling both horizontally and vertically, the instrument has a small thumb-operated button which controls direction of movement. Made by **Smith Drake Corp.**, 1206 S. La Brea Ave., Inglewood, Calif.

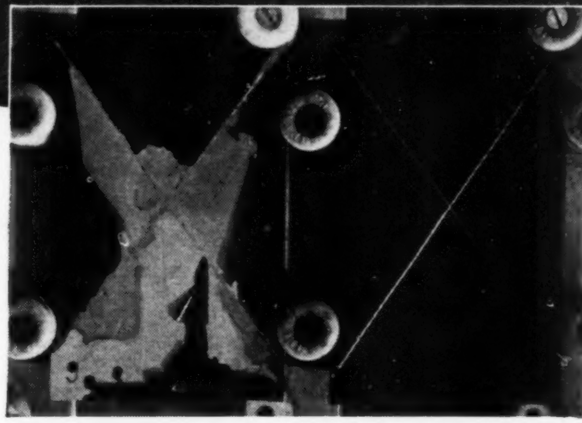
For more data circle MD-140, Page 269

Power Supply

Magnivolt high performance ac to dc regulated selenium rectifier power supply has magnetic amplifier control. Regulation exceeds ±1 per cent from no load to full load with ±10 per cent ac line variation. Response is faster than 0.2-second under extreme load conditions, and ripple is less than 1 per cent rms. Dc output ratings

Parker Rust Proof Company's corrosion test field at Morenci, Michigan.





What weather does to finishes! Panel left, paint over untreated aluminum. Panel right, paint over Bonderite-treated aluminum. 11 months outdoor exposure.

Birds-eye view of a strange crop

A crop of 6,400 metal panels is tended in Parker Rust Proof's outdoor test field. The harvest is facts on the durability of various finishes over various metals.

Conditions here are the same as are met by painted products in daily outdoor use. The sun and the wind, the rain, sleet and snow pound at these panels continually. Parker technicians watch each set of panels individually, observe their condition, chart results at regular intervals.

This large-scale outdoor testing program, continuous since 1935, has helped build Parker's great fund of valuable information on how to control corrosion. The facts we learn from the corrosion test field are passed on to guide our customers to more effective, more economical surface treatments for metals.

**Bonderite, Bonderlube, Parco, Parco Lubrite—Reg. U.S. Pat. Off.*

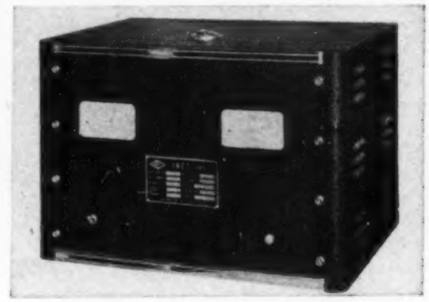
Since 1915—leader in the field

PARKER RUST PROOF COMPANY

2193 E. Milwaukee Ave., Detroit 11, Michigan

BONDERITE	BONDERITE and BONDERLUBE	PARCO COMPOUND	PARCO LUBRITE
corrosion resistant paint base	aids in cold forming of metals	rust resistant	wear resistant for friction surfaces

are 1.2 to 30 v dc and 2.5 to 40 amp full loads from a 115-v, single phase, 60-cycle ac input. Compact and lightweight, the unit requires minimum space for mounting in standard relay racks or standard

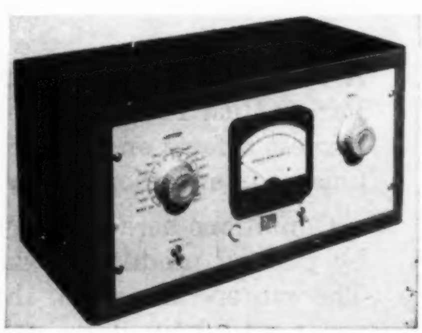


cabinets. It has standard ac input switch, fuse, pilot light, input and output posts. All static components eliminate vacuum tubes or other moving parts. Made by **Inet Inc.**, 8655 S. Main St., Los Angeles 3, Calif.

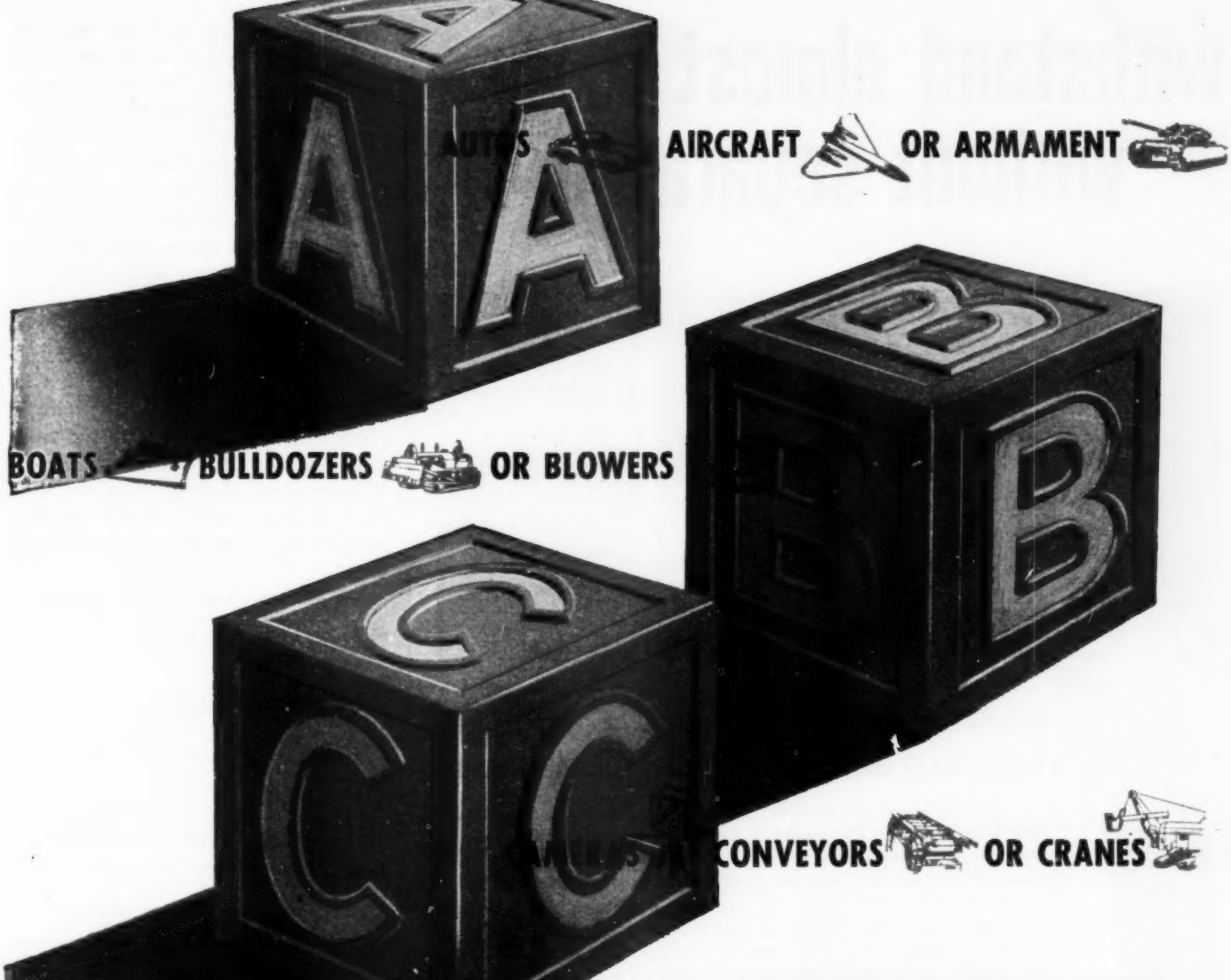
For more data circle MD-141, Page 269

Micro-Microammeters

Vibrating reed and logarithmic scale micro-microammeters measure, record and control microcurrents on the order of one-millionth of a millionth of an ampere. They measure the effect of radioactivity on ion chambers or similar sensing



devices. Basic unit of the model V is the plug-in vibrating reed Vibrode, which is sealed in a glass envelope and can be replaced as easily as a vacuum tube. The instrument has 13 operating current ranges, including 3×10^{-7} and 3×10^{-13} amp full scale. It features



CAPABLE COMPONENTS OF ALL FROM A TO Z

Aetna Bearings and Precision Parts are helping equipment do a better job in almost any field you can name. Reason is, designers, engineers, and manufacturers know that nothing contributes more to equipment efficiency and servicability than quality anti-friction components.

Aetna cooperative engineering has successfully solved over 2000 special anti-friction problems for American manufacturers. Tackling such problems and designing special bearings or precision parts to solve them has been a basic part of Aetna's business for more than a third of a century.

As an executive it may pay you handsomely to find out what Aetna products can do to help make your product more useful, easier to sell, or cheaper to make. A letter, wire or 'phone call will place our engineers at your disposal.

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absolute accuracy of 5 per cent or better on any scale, with less than 2 mv zero drift in 24 hours. Readings are independent of line voltage fluctuations from 103 to 107 v, and relatively insensitive to vibrations. It can be adapted to monitor four inputs alternately and has terminals for connection to any standard 50-mv potentiometric recorder.

For use where input currents are likely to fluctuate, the logarithmic scale micro-microammeter has a wide current range on a single scale. The scale covers six decades, 10^{-7} to 10^{-13} amp, with each decade graduated in nine subdivisions. Made by **Beckman Instruments Inc.,** South Pasadena 1, Calif.

For more data circle MD-142, Page 269

Precision Projector

Model 44 Art-O-Graph casts bright, sharp images of illustrations, photographs, physical objects or color transparencies directly onto a drawing table. Use-

ful in making tracings, it projects illustrations from postage stamp size to the size of a newspaper page and enlarges or reduces images more than four times. Vertical design permits setting the unit up over a regular drawing table. Constructed of steel, the

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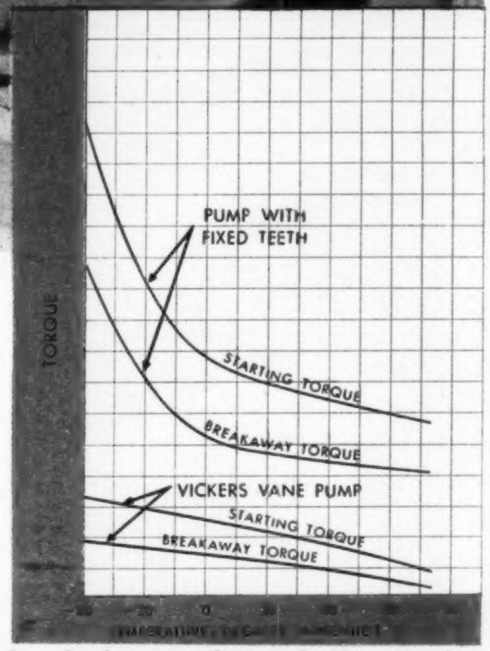
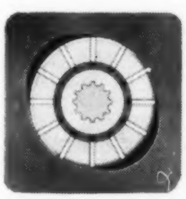
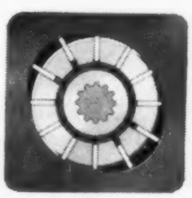


Curves based on comparative tests of a Vickers Vane Type Pump and an equal capacity pump with fixed teeth. Oil used in both was SAE 10W premium grade.



(Left) Schematic diagram of Vickers Balanced Vane Type Pump showing how sliding vanes are retracted at normal engine cranking speeds. No oil is pumped and there is practically no starting load.

(Right) Similar diagram of Vickers pump showing how pump vanes are extended when engine is running. Pumping then begins and continues at all engine speeds.



Vickers Balanced Vane Type Pumps for mobile equipment are available in five basic sizes, 15 normal delivery ratings and a variety of mountings. Ask for new Catalog M-5101.

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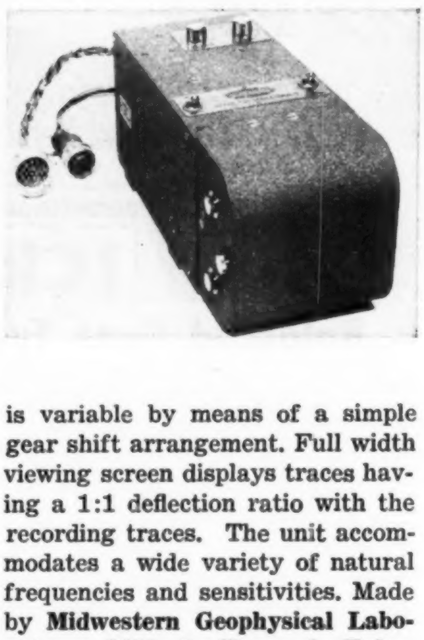
projector has new type foil reflectors, a hard-coated lens, counterbalanced carriage and a front surface mirror. Available from **J. A. Engel Inc.,** 624 Syndicate Bldg., Minneapolis 2, Minn.

For more data circle MD-143, Page 269

Oscillograph

Shock resistant model 557 records 14 channels of separate data, has rigid cast aluminum alloy construction, full width timing lines at 0.01 and 0.1-second produced by a shutter type shockproof precision timer, and beam interrupter type trace identification. Record speed



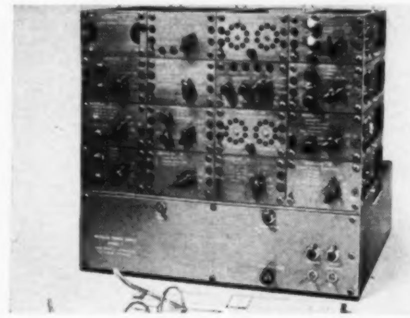
is variable by means of a simple gear shift arrangement. Full width viewing screen displays traces having a 1:1 deflection ratio with the recording traces. The unit accommodates a wide variety of natural frequencies and sensitivities. Made by **Midwestern Geophysical Laboratory,** 3401 S. Harvard, Tulsa, Okla.

For more data circle MD-144, Page 269

Pulse System

Used for transmitting, storing and computing information, the Modular system of digital pulse units consists of 16 electrically and mechanically compatible "Modulars" which perform all the basic functions of digital pulse operations, such as gating, pulse forming, counting, coincidence marking, etc., as well as simpler electronic tasks like amplification, signal in-

version and impedance matching. In operation, the Modulars are easily assembled, linked together by mechanical means and quickly interconnected by patch cords. Since each unit performs numerous

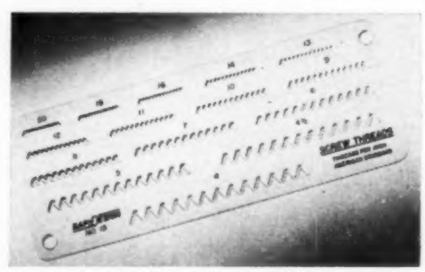


independent functions selectively, a complete system consisting of 16 units and a regulated power supply is capable of 72 separate functions, with as many as 31 functions simultaneously available. Each unit is $2\frac{3}{4}$ in. high, $4\frac{1}{2}$ in. wide and 9 in. long. Made by **Audio Products Corp.**, Los Angeles, Calif.

For more data circle MD-145, Page 269

Screw Thread Template

For drawing the outlines of screw threads, No. 15 template covers the entire range from 20 to 4 threads per inch of the most commonly used American Standard types. This range encompasses

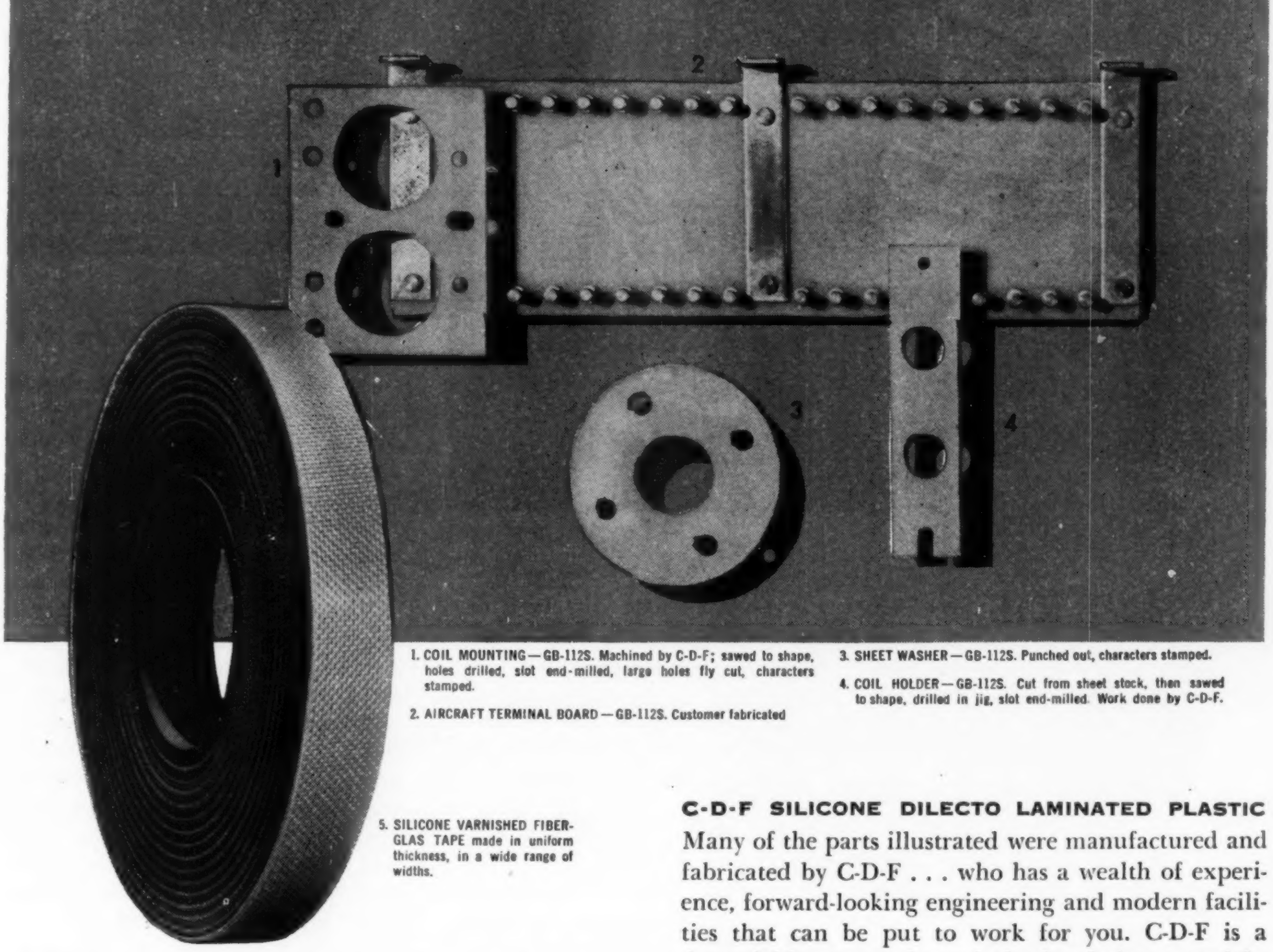


all standard screw diameters from $\frac{1}{4}$ to 4 in. Template is made of cured mathematical quality plastic and measures $6\frac{7}{8}$ by $2\frac{3}{4}$ in. Made by **Rapidesign, Inc.**, P. O. Box 592, Glendale, Calif.

For more data circle MD-146, Page 269

C-D-F SILICONES

For high temperature electrical insulation



1. COIL MOUNTING—GB-112S. Machined by C-D-F; sawed to shape, holes drilled, slot end-milled, large holes fly cut, characters stamped.
2. AIRCRAFT TERMINAL BOARD—GB-112S. Customer fabricated
3. SHEET WASHER—GB-112S. Punched out, characters stamped.
4. COIL HOLDER—GB-112S. Cut from sheet stock, then sawed to shape, drilled in jig, slot end-milled. Work done by C-D-F.
5. SILICONE VARNISHED FIBERGLAS TAPE made in uniform thickness, in a wide range of widths.

C-D-F SILICONE TAPES are recommended for Class H insulation. It's been proved that silicone insulation has 10 times longer life than Class B insulation, even at the temperature limits of Class H. There are two types of C-D-F Silicone Tapes and Sheets: (1) Silicone varnished fiberglas; (2) Silicone rubber fiberglas. Each has the following properties:

- High temperature resistance
- High dielectric strength
- Low dielectric loss
- Resistance to moisture
- High tensile strength
- Flexibility

Both grades meet A.I.E.E. Standard for Class H insulation. They resist mild alkalis, non-oxidizing acids, mineral oils, oxygenated solvents. Silicone rubber fiberglas is recommended for many applications requiring a flexible abrasion-resistant material with good thermal conductivity. C-D-F Silicone tapes and sheets are available in a wide range of sizes in continuous rolls. For complete details, write for Technical Bulletin #47.

C-D-F SILICONE DILECTO LAMINATED PLASTIC

Many of the parts illustrated were manufactured and fabricated by C-D-F . . . who has a wealth of experience, forward-looking engineering and modern facilities that can be put to work for you. C-D-F is a dependable source of supply for insulating materials, and is noted for its fair pricing, for producing high quality products on schedule. Why not call in a C-D-F sales engineer on your problem. Or, write for Technical Bulletins:

#25—complete data on GB-261S, a fiberglas silicone laminate made of a staple filament woven fiberglas cloth and silicone resin in sheet form; #37—covers glass base silicone metal clad laminates; #42—postforming grade of glass base silicone in sheet form; #23—GB-112S, fine weave continuous filament woven fiberglas with silicone resin, sheets, tubes, rods, molded shapes.

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Stress Relief

SOMETIMES a counselor as well, but always a critic, J. P. Henderson resumes his identification of the species—aiming this month at a boss a man could well do without.

Playing Them Close to the Vest

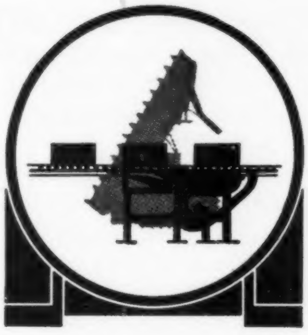
Of all the department heads from whom I would pray to be delivered, the secretive type would appear near the top of the list. He corresponds to the poker player who keeps his hand carefully concealed, unfolding it to the minimum for a quick look with one eye.

You know the type. He never delegates authority. Every decision is made by him. Each problem that comes up, he settles without conferences and discussion. Then he tells everyone exactly what to do. After they have done that he calls them in to tell them the next step. Many times they are completely in the dark as to why they are doing this, or for what broad plan it is a part.

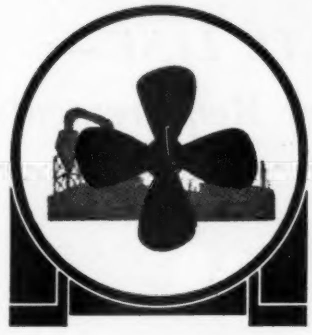
Another version of the same type (and I knew one like this very well) is the top-secret technical man. Somewhere in his desk he has some curve sheets, charts and formulas. Whenever a certain type of technical problem appears it is to be brought to him for solution. He gives the engineer the answer and the engineer proceeds from there.

What an attitude; what a way to run a department! Many times this same type is the loudest in deploring what has happened to people these days. They don't pitch in and work like they used to! They don't take responsibility and he is to be the most pitied because he must work so hard and do all of these things himself since no one under him can handle the problems.

Perhaps his draftsmen and engi-



Material Conveying



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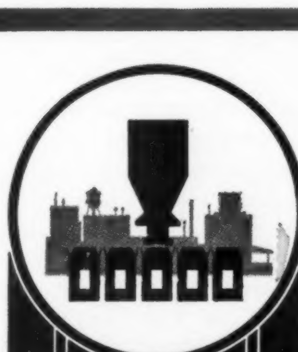


Food Processing

Metal Cutting

Textile Manufacturing

Stress Relief

neers take responsibility; but if so (granted they were of average ability to start) it is obviously because such a boss never gave them a chance to develop.

This type of boss limits growth, not only in his men but for his organization as a whole. I once knew a branch factory with a general manager with this mentality. The plant operated with seeming success for years (although at what cost to the under executives and other personnel would be difficult to determine). There came a time when a sharp expansion was required in the plant and its various supervisory departments. The whole organization split apart at the seams. Now the number of decisions became too great for this man to make all by himself and his men were not trained to fend for themselves. The whole mess had to be reorganized.

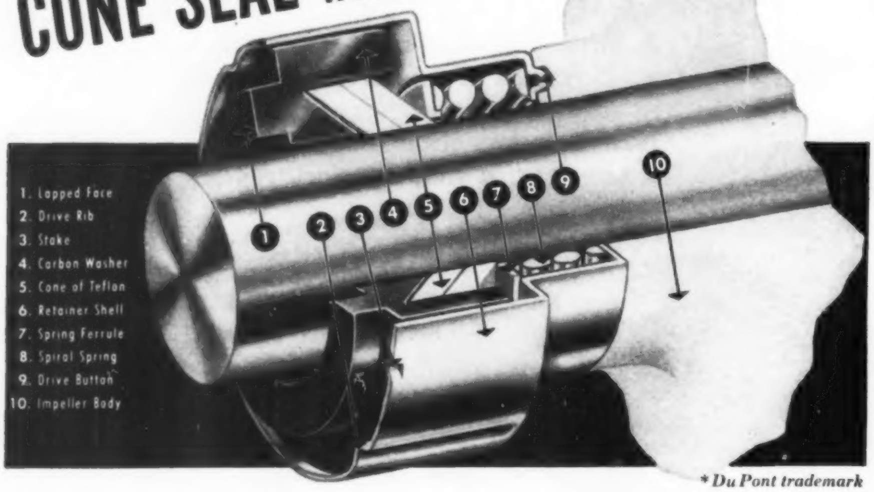
Consciously or unconsciously this attitude often stems from fear. The boss is too afraid of himself, too afraid of his own position to chance developing a rival for his job.

I know of no cure. You just have to outlive this type.

—J. P. Henderson

They Say . . .

"One of the first things any newcomer to industry must learn is the importance of human relations —living and working with people. The engineer particularly must avoid the tendency toward overspecialization. He should seek to develop himself as a whole man, with a broad knowledge and a diversity of interests. The engineer who looks upon the rest of the company as if through the wrong end of a telescope is imparing his own usefulness and prestige, as well as denying himself his rightful place. The creative individual's potential attainment lies within the man himself, and his economic worth will be governed mainly by how well he can express himself as an individual." — Albert F. Watters, *vice-president in charge of personnel, RCA Victor Div., Radio Corp. of America.*



Meetings

AND EXPOSITIONS

Dec. 13-16—

American Institute of Chemical Engineers. Annual meeting to be held at Jefferson Hotel, St. Louis, Mo. Additional information may be obtained from society headquarters, 50 East 41st St., New York 17, N. Y.

Jan. 11-15—

Society of Automotive Engineers. Annual meeting to be held at the Sheraton-Cadillac Hotel and Hotel Statler, Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Jan. 18-22—

American Institute of Electrical Engineers. Winter general meeting to be held at the Statler Hotel, New York, N. Y. Additional information may be obtained from society headquarters, 33 West 39th St., New York, N. Y.

Jan. 22—

Malleable Founders' Society. General society meeting to be held at Hotel Cleveland, Cleveland, O. Additional information may be obtained from society headquarters, 1800 Union Commerce Bldg., Cleveland, O.

Jan. 25-27—

Plant Maintenance & Engineering Conference to be held at the Hotel Conrad Hilton, Chicago, Ill. Additional information may be obtained from the exposition management, Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 25-29—

Institute of the Aeronautical Sciences. Twenty-second annual

Meetings and Expositions

meeting to be held at Hotel Astor, New York, N. Y. Additional information may be obtained from society headquarters, 2 East 64th St., New York 21, N. Y.

Jan 27-29—

Society of Plastics Engineers. Tenth annual technical conference to be held at the Royal York Hotel, Toronto, Ontario, Canada. Additional information may be obtained from society headquarters, 513 Security Bank Bldg., Athens, O.

Feb. 3-5—

Society of Plastics Industry. Ninth annual reinforced plastics division conference to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 67 West 44th St., New York 18, N. Y.

Feb. 15-17—

American Management Association. Personnel conference to be held at the Palmer House, Chicago, Ill. Additional information may be obtained from society headquarters, 330 West 42nd St., New York 18, N. Y.

Mar. 2-4—

Society of Automotive Engineers. National passenger car, body and materials meeting to be held at Hotel Statler, Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Mar. 15-19—

National Association of Corrosion Engineers. Tenth annual conference and exhibition to be held at the Kansas City Municipal Auditorium. Additional information may be obtained from society headquarters, 1061 M & M Bldg., Houston 2, Tex.

Designing Hydraulic Equipment

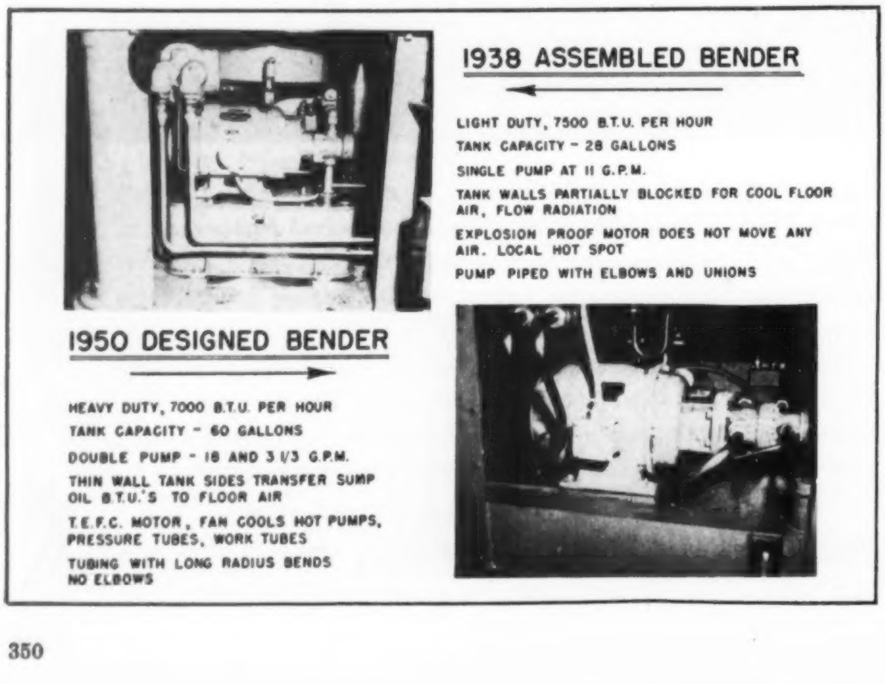
. . . for heat dissipation

By J. Russell Hemeon
Hydraulic Equipment Engineer
Ternstedt Div.
General Motors Corp.
Trenton, N. J.

IT IS the responsibility of the designer to be fully cognizant of the heat problem involved in the design and performance of hydraulic equipment. The assembly of pumps, valves, and piping without careful analysis of anticipated pressure may develop an excessive sump oil temperature. Time cycle of operation, Btu's per hour generated, and balancing capacity for heat dissipation are factors that enter the picture. Tank capacity and its design are important. Rates of cooling from air flow over thin-wall steel tubing, fin coolers, chamber capacity, and water coolers must be evaluated. The end result is to provide an economic heat dissipation system which will insure a satisfactory sump oil temperature. The dividend is increased oil and pump life with extended periods between preventive maintenance overhauls.

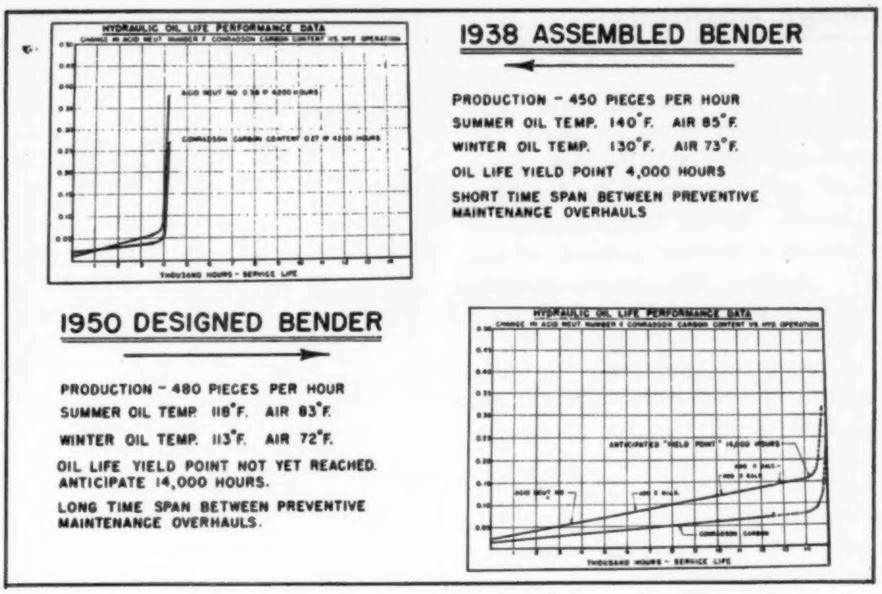
Assembling vs Designing: An early model of a bender is shown in the upper half of *Fig.* 1. This bender was simply assembled with little thought given to heat flow efficiency. Contrast the limitations of this unit with developments made in the designed bender shown in the lower half of *Fig.* 1. This designed unit had good control of heat generation and higher heat dissipation capacity. The trend of oil life improvement with the designed bender is depicted in *Fig.* 2. These curves show the hydraulic oil "yield point" at 4,000 hours on 1938 assembly, against an anticipated 14,000 hours for 1950 design. Oil life yield point is similar to its cognomen in steel. Oil life yield point is defined as the sharp change from a constant rate of increase in the value of the Conradson carbon content and acid neutralization number. After abuse of hydraulic oil, particularly by operating at an elevated sump temperature, oxidation increases especially from high temperature absorbed air. This results in a molecular breakdown with an attendant lack of lubricity. When operating at 140 F, the yield point is reached at 4000 hours as shown in the upper half of *Fig.* 2. Design which limited oil temperature to about 115 F increased oil life yield point to 14,000 hours. To achieve results a careful estimate of the expected generated heat must be made, and then the equipment must be designed with sufficient heat dissipation to obtain a desirable sump oil temperature.

Fig. 1—Comparison of poorly designed bender, upper half, and well designed bender, lower half



Heat Generation: The best method to evaluate the factor of heat generation is to compute the pump output in terms of Btu per hour.

Fig. 2—Right—Hydraulic oil life comparison data of designs in *Fig.* 1



In *Fig.* 3 the development is shown for a basic formula which converts psi pressure into feet of head, and actual gallons per minute into pounds per minute. Note that the formula, Btu per hour $= K \times 1.482 \times$ psi $\times$ gpm contains a K factor. This is the factor for pump efficiency and percentage of air in the oil. Well designed 1953 equipment employs a K factor value of 1.05.

The poorer the design the higher the K factor; some have been investigated that had a value as high as 1.8. One of the worst offenders for raising this K factor and thus generated Btu is too high a percentage of air entrapped in the oil. It is not sufficient to simply specify an antifoaming agent in the oil. All return tubing must be submerged and oriented to obtain the desirable long-flow heat transfer and air escape potential. Intake tubing to the pump must be made of one piece and tight to eliminate air suction leaks. That percentage of leaked or entrapped air, admixed with oil, when compressed to 600 psi will have a theoretical air temperature in the range of 1000 F. It is recognized that the percentage volume of air in oil is small, but it has been found that excess entrapped air has raised the sump oil temperature as much as 25 per cent.

The hot compressed air molecule does fry the microscopic film of oil surrounding the air and thus gradually hastens the yield point breakdown. Undue restrictions, undersize control valves and fittings, with high pressure drop losses, small tubing with a high Reynolds number are also sources of unnecessary heat generation. These detrimental factors must be eliminated or limited to minimize the value of K.

Pressure vs. Time Curve: To determine the mep (mean effective pressure) for use in the Btu generation conversion formula a pressure-time curve is necessary. The best way to anticipate a curve for a proposed design is to make an accurate plot of production opera-

Fig. 3—Below—Btu generation conversion example

PROJECT: TO SET UP THE DATA TO BE USED IN A FORMULA WHICH WILL CONVERT PUMP PRESSURE (P.S.I.) AND PUMP VOLUME (G.P.M.) INTO RATE OF HEAT GENERATION (B.T.U. PER HOUR). THIS IS A FUNDAMENTAL STEP FOR THE DETERMINATION OF TANK SIZE AND THE CORRECT SELECTION OF RADIATION PROVISION AND COOLING REQUIREMENTS THAT WILL RESULT IN THE DESIRABLE SUMP OIL TEMPERATURE OF 115°F.

CONVERT P.S.I. TO PRESSURE HEAD IN FEET

144 ÷ 62.43 X 0.878 (AVG. SP. GR. HYD. OIL) = 2.621 X P.S.I. = FT. OF OIL HEAD

CONVERT G.P.M. TO POUNDS PER MINUTE

8.3457 LBS. OF WATER/GAL. X 0.878 = 7.3275 LBS. OF OIL/GAL.

WORK – FOOT POUNDS

$$\frac{2.621 \times 7.3275}{777.5} \times 60 = 1.482$$

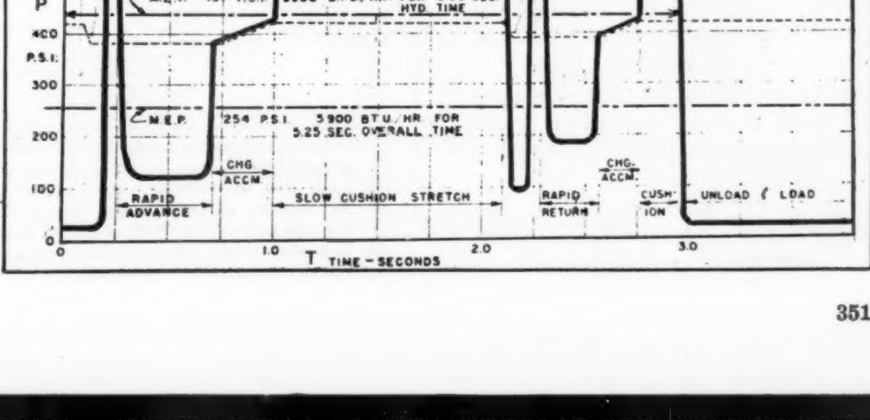
∴ B.T.U. PER HOUR = K × 1.482 × P.S.I. × G.P.M.

K = FACTOR FOR PUMP EFFICIENCY AND PERCENTAGE AIR IN OIL
K = 1.05 FOR AVERAGE INSTALLATIONS

EXAMPLE: THE STUDY OF A TYPICAL P.T.S. DIAGRAM SHOWS A M.E.P. (MEAN EFFECTIVE PRESSURE) OF 250 P.S.I. WITH A PUMP RATED AT 11 G.P.M. WHAT IS THE B.T.U/HOUR?

B.T.U./HOUR = K X 1.482 X P.S.I. X G.P.M.
= 1.05 X 1.482 X 250 X 11.0
= 4279.3

Fig. 4—Below—Final pressure-time-solenoid hot chart for determining optimum psi and time for minimum heat generation for typical stretch bender



tion for each new machine. The desired force expected from the hydraulic equipment must be translated into an anticipated working pressure. Initial design should be based on a working pressure of 500 psi. A final PTS (pressure-time-solenoid hot) chart for a typical stretch bender is shown in *Fig.* 4. For use in the conversion formula, integration under the curve shows an mep of 254 psi for an overall time of 5.25 seconds. This converts to a theoretical 5900 Btu per hour with K equal to 1.1. This curve was made after final optimum adjustments.

It is common practice to set the pump relief valve 20 per cent above the peak demand. On this bender, there was a ram return inertia starting demand of 800 psi. This would establish the relief valve at 1000 psi. In consequence, the slow cushioned stretch, dumping excess pump volume over the relief valve would have resulted in an mep of 398 psi and a corresponding 9100 Btu per hour. "Decapitation" of the relief valve setting to 700 psi as shown, cushioned the start of the ram return and materially reduced the area under the pressure curve where the slow cushion stretch was making heat at the relief valve setting.

There is an upper mep of 437 psi in *Fig.* 4 for the hydraulic time cycle of 2.95 seconds. This is equivalent to 9900 Btu per hour. The ratio of hydraulic time Btu to overhaul time Btu at 1.68 is above a critical 1.5 to 1 ratio and requires special treatment with extra heat dissipating capacity.

In aircraft, parasitic drag is eliminated or reduced in airfoil flow design. To obtain top hydraulic performance, parasites must be recognized and similarly eliminated. One of the worst parasitic drags comes from cheap unstable oil which does not possess long life stability and lubricity. It is of the utmost importance to specify a quality hydraulic oil that will meet the limitations of individual units. Low grade oil may cause trouble in

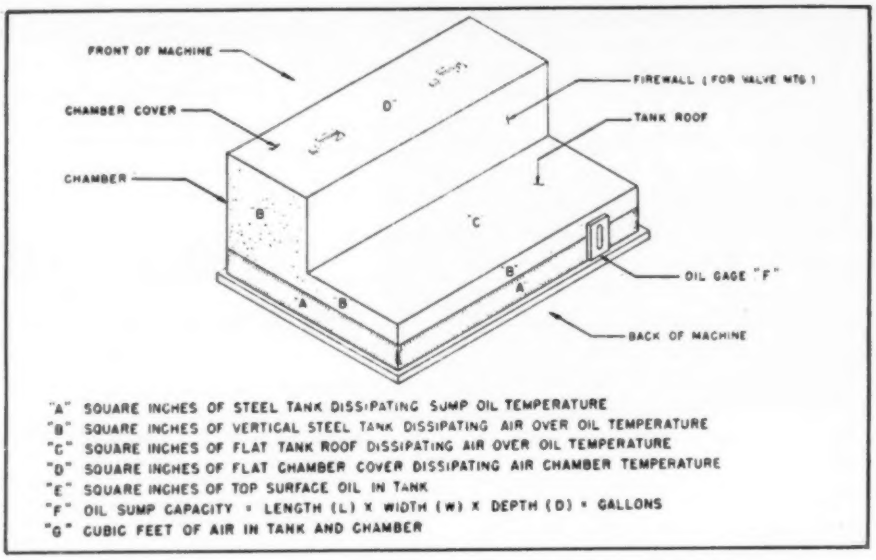


Fig. 5—Above—Oil tank design for efficient heat dissipation

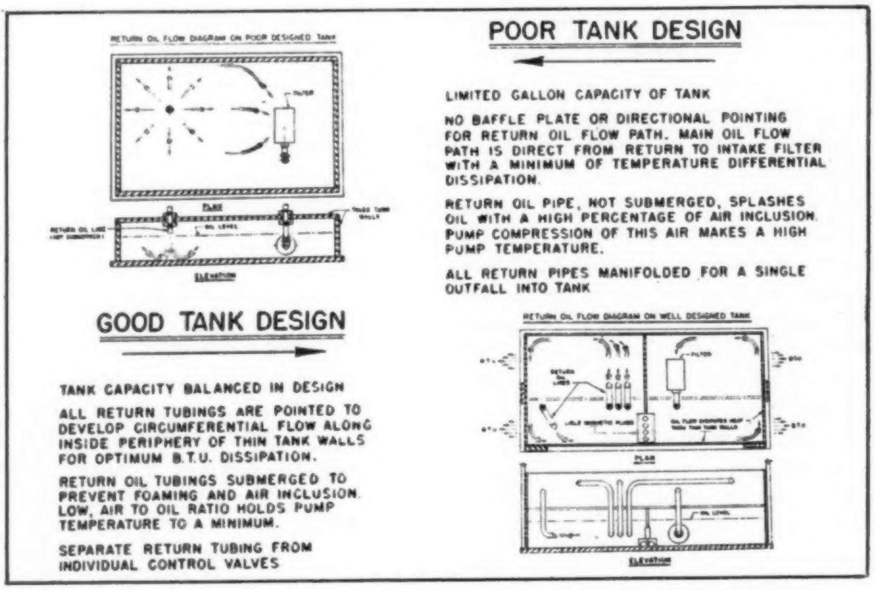


Fig. 6—Above—Comparative features of poor and good tank design

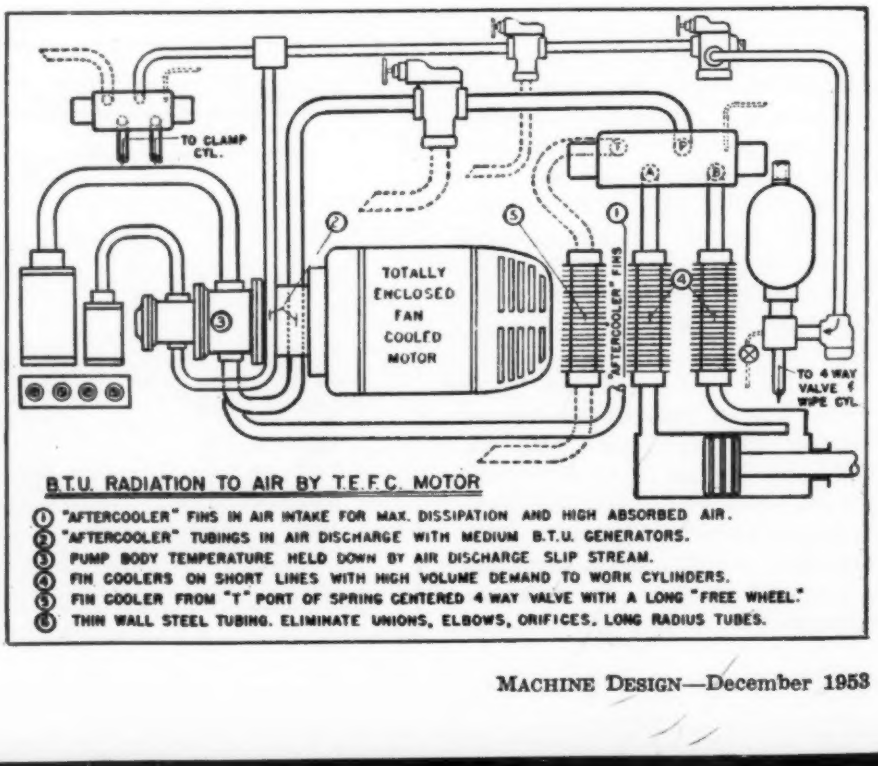


Fig. 7—Right—Five effective methods for air cooling hot oil

less than 1000 hours. This is evidenced by sludging and starving intake filters, scoring, varnishing and pitting pump and valve parts.

Another source of drag is insufficient lubrication. Design must incorporate adequate accessible oil and grease fittings. The lubrication engineer must set up that correct periodic schedule for hours of satisfactory performance between servicing, with the correct type and quality of lubricant. Automatic lubrication is now a must on major units.

Particular emphasis is placed on the necessity for full floating hydraulic cylinders to eliminate misalignment loads from a sprung piston rod on the stuffing box seal. Friction makes heat and calls for more pressure.

Reduce the Btu generation in design, and then evaluate the amount of heat to be made and select the correct dissipation potential by radiation, fan cooling, and water cooling to make sure that economic performance will result.

Fig. 8—Below—Comparison of laminar and turbulent flow

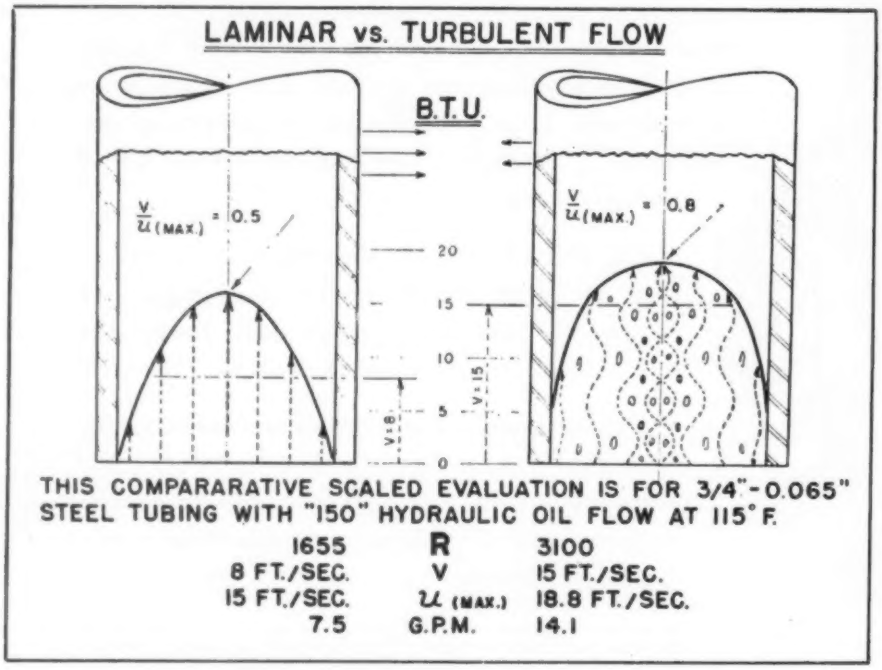
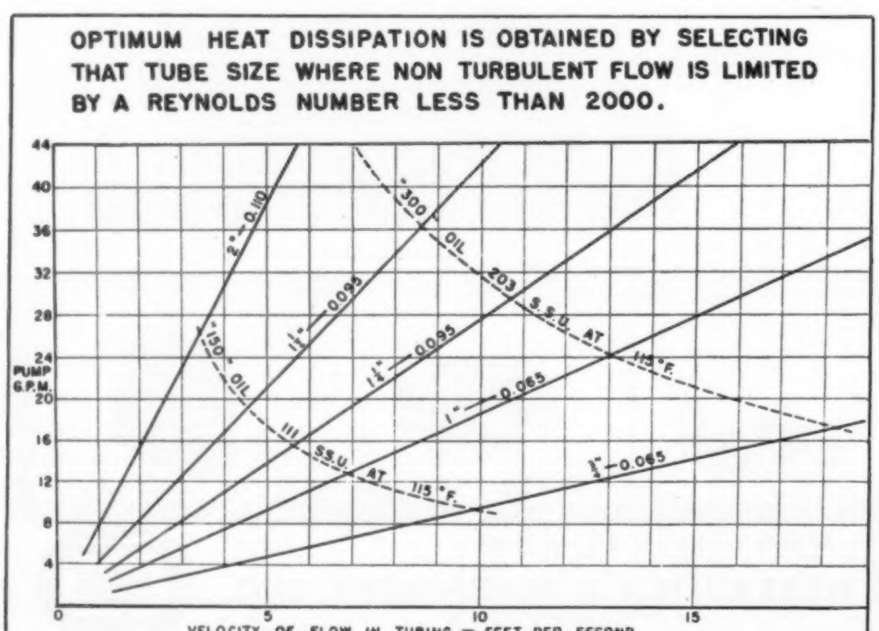


Fig. 9—Below—Chart to insure nonturbulent flow in hydraulic equipment



Radiation Cooling: Radiation shall be construed to encompass the provision for the tank to transfer heat from the return oil through the thin tank wall to the cool floor-level air, this radiation to be further augmented by the transfer of hot-chamber air through the chamber cover and walls.

A successful type of tank chamber design to handle up to 10,000 Btu per hour is illustrated by *Fig.* 5. The various control factors A to G must be studied and balanced to come within floor space restrictions for the best radiation efficiency.

Careful preparation of scale layout drawings results in time saved on the erection floor. Also, a scale layout helps the designer think the job through and eliminates elbows and pockets that will make hot spots.

The essential features of poor and good tank design are shown in *Fig.* 6. The gallon capacity of the tank may vary between 3 and 5 times the rated pump capacity. Key pointers for successful design are given in *Fig.* 6 and should be studied with care. In this design it was permissible to use 1/4-inch steel plate tank walls for a rapid rate of heat transfer. Tool, die, and table top load were supported on 1 by 6 inch vertical steel pilasters.

Notice in the good tank design that the mid-tank baffle does not extend across the full width of the tank. Toward the forward wall of the tank is a four-holer Lisle magnetic plug assembly. This unit magnetically holds the micron fine particles of abrasive matter that would otherwise pass through the suction filter and act as a lapping compound which will affect the life of the pump cartridge. Magnetic plug assembly is under the chamber cover, permitting easy removal for cleaning while the machine is in operation.

The suction filter is also accessible through the chamber cover and is made up hand tight on a short nipple for service replacement without shut down. Too many designs bury the filters with the

result that they are not serviced. A clogged filter creates a semivacuum in the pump intake system to produce a pump "anvil chorus". The pseudomechanic fixes the job good by removing once and for all the clogged filter so the pump will not knock again.

Fan Cooling: The totally-enclosed, fan-cooled pump motor and the arrangement of thin wall steel tubing and fin coolers can be effectively employed in the influent and effluent air stream to dissipate heat economically from the hot oil flow in tubing.

In *Fig.* 7 a schematic hydraulic flow diagram is presented using two pumps and an accumulator to power a three-function circuit. This diagram shows five possible methods of motor fan cooling to dissipate heat for attaining the desired stabilized sump oil temperature.

Circle 1 shows hot pump oil giving up heat by means of an aftercooler fin assembly located at the cold air intake to the motor. This arrangement is a must where 10,000 Btu per hour are generated. Fin aftercooling should also be used where the tank-chamber dimensions are limited to the extent that it is impossible to anticipate a good temperature differential between return oil and intake oil. When peak pressures are in the 1000 psi range and there is not complete protection to minimize the percentage of absorbed air in the oil, this system is advised to effectively and immediately reduce the extreme high temperature of the air molecules.

Circle 2 presents pump aftercooler tubes making an immediate "U" bend into the air discharged from the motor. This is an economic selection for units generating less than 8000 Btu per hour and where the mep does not exceed 450 psi. The large pump tubing should be high and the small pump tubing should be low to realize the best results.

Circle 3 depicts the effluent air stream completely enveloping the pump. This is an inherent characteristic on all totally-enclosed, fan-cooled motor installations. The pumps are the "frying pans" where

Case History M-105

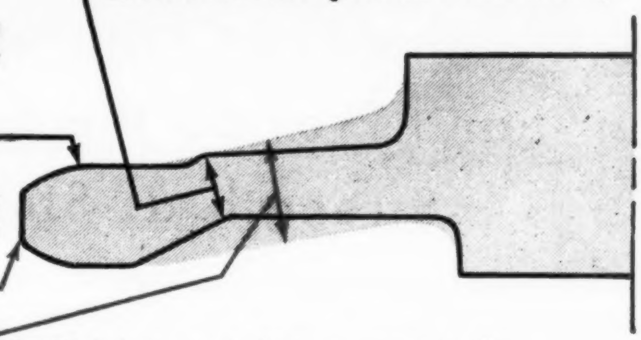
Here's what we mean by SUPERIOR ENGINEERED FOUNDRY PRODUCTS...

PROBLEM:

1. Conventional design of cast steel sprocket blank impaired proper directional solidification of the metal, causing uncontrolled shrinkage in the outer rim.
2. Assurance of delivery of sound castings was possible only through expensive non-destructive inspection of each blank.
3. Casting and machining losses were excessive.

OUR SOLUTION:

FOUNDRY ENGINEERED DESIGN to establish a relationship of tooth, web and hub sections which would insure uniform directional solidification of the metal, place metal where needed most and minimize necessity for expensive inspection of each part.



RESULT: 17.6% SAVINGS

1. Elimination of shrink holes in outer rim through proper directional solidification of the metal.
2. Elimination of need for expensive inspection of each part.
3. Reduced casting and machining costs.

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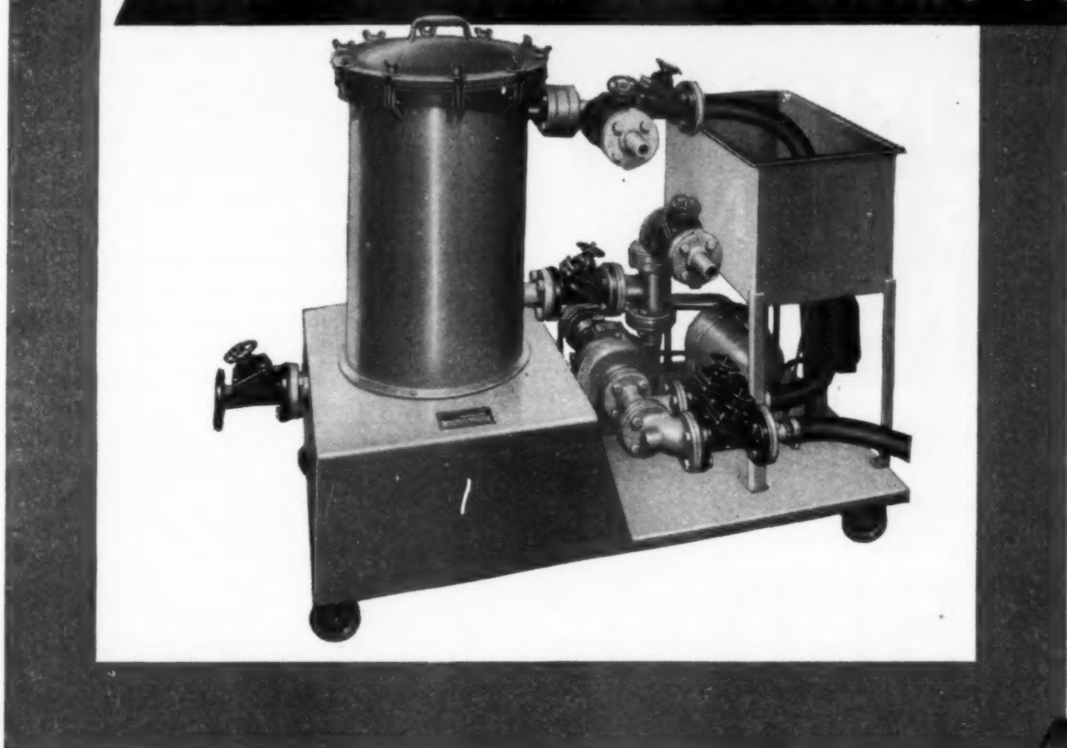
pressure and temperature are produced.

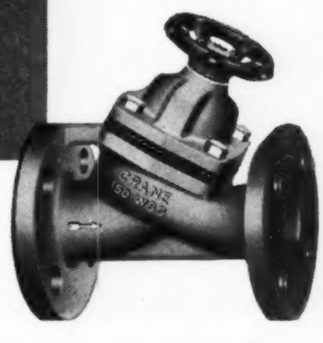
At circle 4 is a pair of aftercooler fins located in the influent air stream. This is not the most economical design and should only be employed as a last resort. Also this arrangement should only be used when the cylinder demand volume is high and the tubing runs are short. If these limitations are not observed, the oil volume will only shuttle back and forth in the fin tube. There is no appreciable oil slug progression to transfer cooled oil to the sump efficiently. On certain installations where the work cylinder is relatively high and consequently forms detrimental air pockets, it has been necessary to use a combination bleeder and check device to insure progression of air molecules entrapped in oil slugs for venting in the tank. This bleeder-check has eliminated soft sponge-like tool advance. If a bleeder-check to the tank is designed into the circuit, the use of an aftercooler fin at point 4 can be justified.

Heavy Duty Cycles

The use of a fin cooler located at the influent air stream to the motor is shown at point 5. This system is highly beneficial when the cycle time is made up of more than 40 per cent of unload and load time. In this time, the pump must "free wheel". To accomplish this free wheeling requires a four way valve that is tandem center or open center and spring centered. Both solenoids must be dead to permit the pump to free wheel full oil volume from the P to T port at no pressure. Hence each volume-time slug of no pressure, no temperature rise oil, is supercooled at the fin cooler and enters the tank at a substantially lower temperature. On heavy duty cycles up to 12,000 Btu per hour, some designs have incorporated an outside location of return oil tubing. With an origin at the T port of the four-way valve, this tubing bulkhead unions through the firewall for a first stage cooling in the effluent air, and then passes outside the motor with long radius bends for a second stage cooling

HOW TO FILTER ACID PLATING SOLUTIONS





another successful product design with piping equipment by CRANE

Belke Manufacturing Co., of Chicago, Ill., has an answer to the problem of filtering acid plating solutions.

In the compact Belke unit, chemically inert Lucite plates inside a filter bag provide large surface for fast, effective filtering. All tanks and piping are rubber lined to nullify the corrosive action of the plating solutions. Finding valves for such severe service was not as difficult as you might imagine—they're Crane rubber-lined diaphragm valves, regularly available.

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in the influent air. The path is through the firewall again for an advantageous submerged and pointed location.

On lighter duty cycles, this design can save the cost of a fin cooler by using large size tubing at a Reynolds number well under 2000. In consequence, first and second stage progressive tube cooling is the epitome of free wheeling Btu transfer.

It is necessary when selecting the motor size not to exceed the recommended horsepower rating. Overloaded motors will have a higher effluent air temperature and a much lower cooling capacity for the conditions presented by circles 2 and 3. In the design and construction of hydraulic equipment, take advantage of T.E.F.C. cooling to reduce the marginal cost of water to auxiliary coolers.

Reynolds Number Limit

The estimable work by Osborne Reynolds is well documented in current textbooks. There is only meager emphasis on limiting some of the variables and making a practical application to a recognition of the Reynolds number, *R*, for obtaining optimum heat dissipation. The ineffective, time-worn rule, "Flow velocity in power tubing should not exceed 15 feet per second," must be discarded as it does not accord recognition to the vital operating viscosity of the oil that is used.

In *Fig.* 8 a comparative scaled evaluation is shown for the flow of No. 150 SSU hydraulic oil through ¾-inch diameter, 0.065-inch wall-thickness steel tubing. At 115 F the viscosity is 111 SSU. As shown on the left side of *Fig.* 8 a 7.5 gpm pump will develop a theoretical mean flow velocity of 8 feet per second. In the true analysis, the velocity flow has vectors with a maximum flow, $u_{max} = 15$ feet per second of the polar center. Under these conditions with the Reynolds number of 1655, the flow is laminar or streamline and good heat transfer will be provided. On the right side, a 14.1 gpm pump develops the old accepted mean flow $V = 15$ feet per second. The u_{max}

is 18.8 feet per second. The Reynolds number is 3100. The velocity vectors are turbulent and the Btu is materially reduced.

Current design practice requires a selection of tubing size that does not permit the Reynolds number to exceed 2000. In a case history example, a hydraulic machine was using combined power and free wheeling cooling with ¾-inch tubing. The tubing had a strong finger-touch vibration and a Reynolds number of 2700. The stabilized sump oil was at 121 F or 100 SSU. The only design change was an increase of the two-stage tube cooler to 1 inch tubing. Vibration was completely eliminated, the Reynolds number computed to be 1550. The sump oil temperature was reduced and stabilized at 111 F or 120 SSU.

It is advisable to design for an optimum sump oil temperature of 115 F where the operating viscosities are 111 and 203 SSU respectively. Hot operating, oil temperature thins out the viscosity and increases the Reynolds number. When R is above the critical value with turbulent oil flow, the heat dissipating potential is reduced and an inflationary spiral is in progress.

Reynolds number, critical velocity, viscosity, and heat dissipation must all be correlated for optimum design performance.

A rough analysis of a hot sump oil tank or a vibrating pipe may be made by using the chart in *Fig.* 9. Locate the particular pump gpm on the ordinate and move to the right until the tube size diagonal is intersected. This intersection probably will be to the right of the dotted oil curve if the oil flow is in turbulence.

Water Cooling: On high duty cycles of operation where there is insufficient radiation and fan cooling capacity to balance the generated heat, it becomes necessary to design water cooling in. For example a piece of die casting equipment with a K of 1.5 generated 33,000 Btu per hour and required 7 gpm cooling water. The equipment was redesigned which resulted in the reduction of K to 1.2. The

unit only generated 26,000 Btu per hour and required but 0.9 gpm cooling water.

The new design had more efficient radiation characteristics, too. A close-up view of a heat exchanger system on the side of the tank of this die cast machine is shown in *Fig.* 10. The fully automatic system had a high initial cost which was justified by operating economies and freedom from maintenance problems. The path of the influent cooling water starting from the extreme right goes through the following steps:

1. Screen sediment filter to trap rust and scale.
2. Solenoid controlled shut-off valve wired across the motor pump.
3. Water flow control modulating valve with bulb in the tank, to control cooling water flow to maintain 115 F sump oil.
4. Heavy duty highly efficient four pass heat exchanger.

Oil flow to and from the cooler is provided by means of steel tubing in place of the usual pipe, elbows, unions, and nipples.

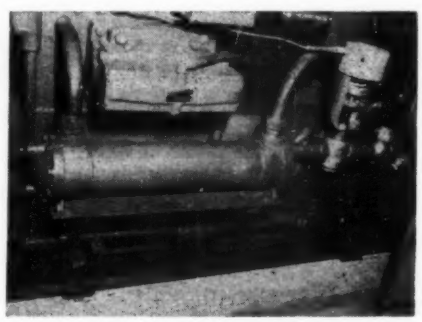

Fig. 10—Die-cast machine with fully automatic water cooler

Uncontrolled temperature can be disastrous in causing machine failure and frequent repair calls. The compilation of service failure data, improvement of low life expectancy units, and the design of hydraulic equipment for heat dissipation for a sump oil temperature of 115 F can be large contributing factors in extending the satisfactory hours of operation between scheduled overhauls.

From paper entitled "Design of Hydraulic Equipment for Heat Dissipation" presented at Annual

Design Abstracts

Meeting of ASME in New York, N. Y., December, 1953.

High-Temperature Brazing

By George D. Cremer, Frank J. Filippi and Richard S. Mueller

Research Div.
Solar Aircraft Co.
San Diego, Calif.

HIGH-STRENGTH joints comparable to base metal continuity can be achieved by various welding techniques. Through careful design, brazed joints with similar physical properties may also be realized. Whereas sound welds generally exhibit the properties of the base metal, brazed joints by their very nature of application usually are limited to considerably lower operating temperatures. Certain rapidly diffusing brazing alloys provide interesting exceptions to this generalization.

From an oxidation resistant and chemical corrosion standpoint, the choice of suitable brazing materials is sharply limited as compared to welding of a wide variety of suitable base metals. As is well known, the welding process also has its limitations and in these areas brazing is frequently the answer.

Brazing is a thermal process for effecting atomic bonding of solid surfaces by means of an intermediate, fused filler metal. *High-temperature brazing* is a metallurgical joining process in which a molten filler metal having a melting point in excess of 1600 F but below that of the base metal is used. Diffusion between filler metal and base metal must occur.

The more common commercially available brazing alloys are listed below. Copper is included for comparative purposes only, as it is seldom used in high-temperature, corrosion-resistant applications. A possible exception might be short-lived applications, such as rocket components.

1. Copper
2. Coast metals
3. Silver-manganese
4. Manganese-nickel

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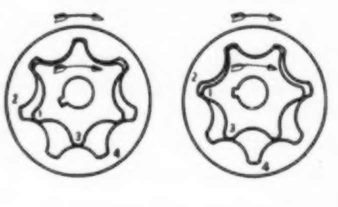
The two Gerotors revolve at different speeds, resulting in a continuous opening and closing of spaces as shown in this diagram. Follow the changing position of tooth #1 as rotation takes place. Note its relation to space #2. As rotation continues, tooth #1 gradually displaces oil from space #2, thus creating pressure.



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5. Palladium-silver
6. Palladium-nickel
7. Solabraze
8. Nicrobraz

A number of other high temperature brazing alloys are known to be in limited usage or in developmental stages. Nickel-chromium - silicon, nickel - germanium, nickel-silicon-phosphorus, and nickel-chromium-phosphorus alloys deserve mention.

Copper: In copper brazing the braze metal as well as the base metal, which is generally mild or low alloy steel, is usually not corrosion resistant. Copper brazing is a high temperature operation with furnace temperatures of 2000 to 2150 F required. Careful control of temperature, time, and quality of the reducing atmosphere is essential. The process has been highly developed as a precise metallurgical production tool. Many thousands of tons of successfully copper-brazed parts have firmly established its production significance and economies. Intricate, multicomponent assemblies can be copper brazed using completely automatic equipment.

Coast Metals: A series of brazing alloys have been developed by *Coast Metals Inc.*, with compositions which contain a minimum of 85 per cent nickel, 2-3 per cent boron, and 3-5 per cent silicon. One alloy contains in addition approximately 7 per cent chromium. Major characteristics include:

1. Flow point: 1850 F.
2. Recommended furnace temperature: 1900 to 2050 F.
3. Brazing done with flux or in protective atmosphere.
4. Available in powder or cast rod form.

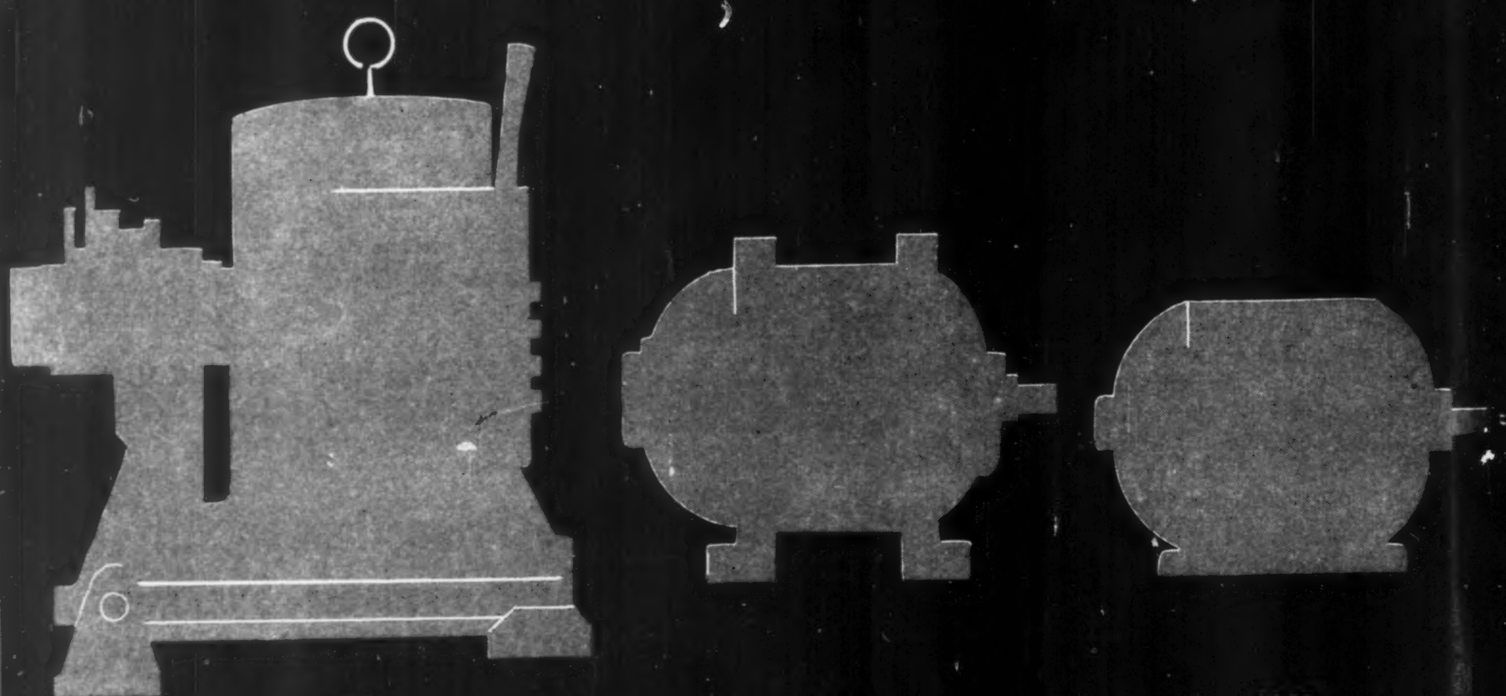
Silver - Manganese: Application of this brazing material includes joining of stainless steel and high-alloy parts for elevated temperature use up to about 1300 F. Manufactured by *Handy and Harman*, its characteristics are the following:

1. 80 Ag, 15 Mn
2. Tough and ductile, solid-solution type alloy

(Continued on Page 370)

WHAT'S NEW

IN MOTORS



1888	1926	1947
TESLA	CS	LIFE-LINE
5 H.P.	5 H.P.	5 H.P.
765 LBS.	130 LBS.	119 LBS.

Design Abstracts

(*Continued from Page 367*)

3. Melting point: 1750 F
4. Flow point: 1760 F
5. Furnace temperature required: 1850 to 1950 F
6. Available in wire or strip form
7. Brazing done with "H" flux or pure, dry hydrogen, −80 F dew-point

Successful brazing applications using Ag-Mn include attachment of Inconel fuel line fittings, joining of Stellite valve seats to stainless valve bodies, and applying of cemented carbide tool tips to steel shanks for hot machining operations.

Manganese-Nickel: Also made by *Handy and Harman,* brazing conditions for this metal are similar to those of the preceding silver-manganese alloy. Its main characteristics are:

1. 68 Mn, 32 Ni
2. Tough, solid-solution type alloy
3. Melting point: 1850 F
4. Flow point: 1860 F
5. Subject to scaling at elevated temperatures
6. Available in wire or powder form
7. Operating temperatures range up to about 1500 F

Since this alloy is not attacked by molten sodium, a potential use is in liquid cooled nuclear reactors.

Palladium-Silver: Produced by the *Mond Nickel Company Ltd.* and *American Platinum Works,* the following characteristics for the alloy are known:

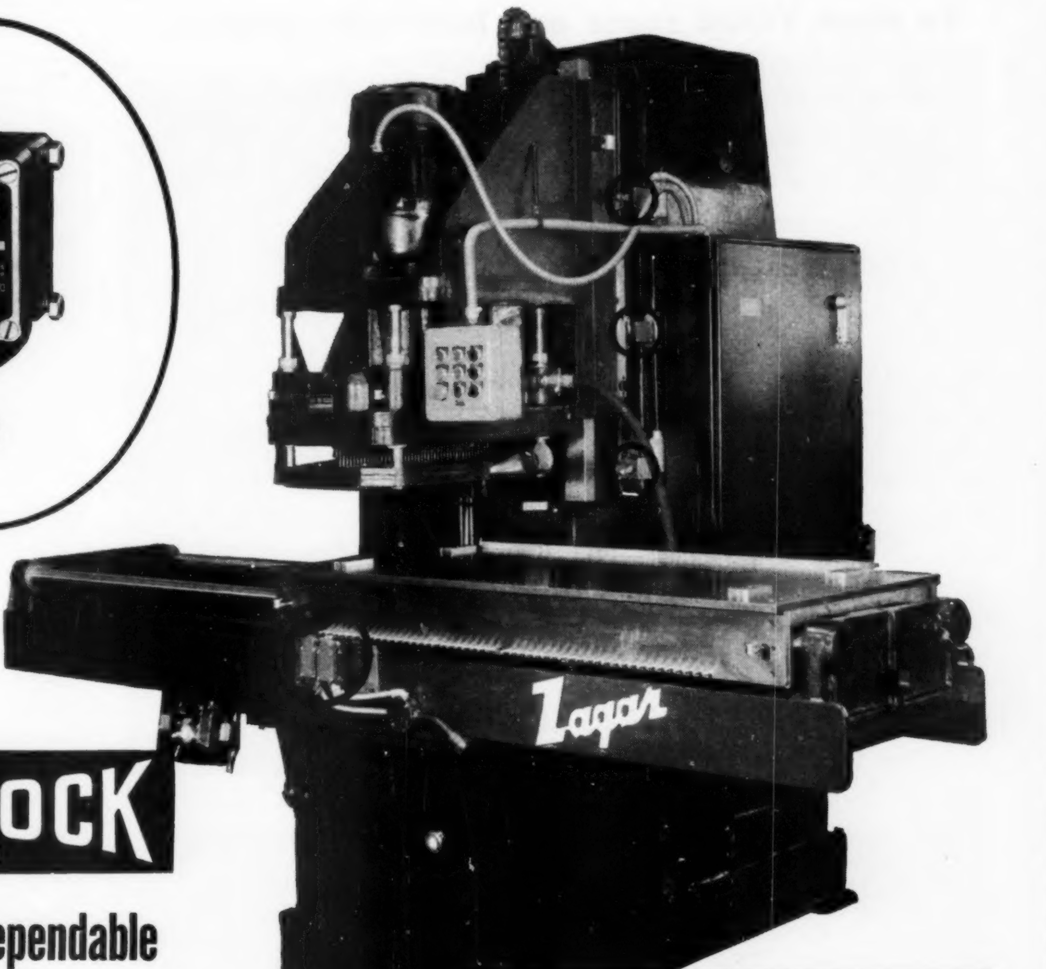
1. 75 Ag, 20 Pd, 5 Mn
2. Tough, ductile alloy
3. Brazing temperature: 2050 F
4. Available in wire or strip form

Only a limited amount of information is available concerning this alloy. Another similar high temperature brazing alloy has a recommended brazing temperature of 2240 F.

Palladium-Nickel: Produced by the metallurgy division of the *National Laboratory* at Oak Ridge, this experimental alloy has the following characteristics:

1. 60 Pd, 40 Ni
2. Tough and ductile, solid-solution type alloy
3. Flow point: approximately 2250 F

for jobs where <u>you</u> "can't take chances"...



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Design Abstracts

4. Brazing temperature: 2350 F (approximate)
5. Available in wire, strip, or powder form

Solabraze: A number of alloys are currently being developed by *Solar Aircraft Co.* to meet specific classes of high-temperature brazing applications. Compositional ranges are in general 60 to 85 per cent nickel, up to 20 per cent chromium, up to 30 per cent manganese, ½ to 5 per cent silicon, 0 to 3 per cent boron, together with varying amounts of iron and molybdenum. Main characteristics are:

1. Flow point: 1750 to 1950 F
2. Available in powder and cast-rod form
3. Brazing done with flux or pure, dry hydrogen

A useful characteristic of the brazing process is of great significance in large, multicomponent parts. Overall distortion during brazing can be maintained within remarkably small limits. It is not unreasonable for example, to maintain flatness over a two-foot diameter within 0.030-inch.

Nicrobraz: Produced by the *Wall Colmonoy Corp.* and made in accordance with Aeronautical Material Specification No. 4775, the nominal alloying composition of this nickel-base brazing alloy is 4 per cent silicon, 16.5 per cent chromium, 4 per cent iron, 3.8 per cent boron, and 1 per cent carbon. Its other characteristics are:

1. Hard and highly corrosion resistant
2. Flow point: approximately 1930 F
3. Brazing temperature: 1950 to 2150 F

Nicrobrazed joints can be made which exhibit useful strengths at temperatures in excess of the melting point of the massive brazing alloy itself. With relatively tight fitting brazed joints, the effects due to atomic diffusion at the brazing temperature occur very rapidly. For example, in the case of stainless steel joined with Nicrobraz, mutual diffusion between the base and braze metals produces an entirely new series of alloys in and

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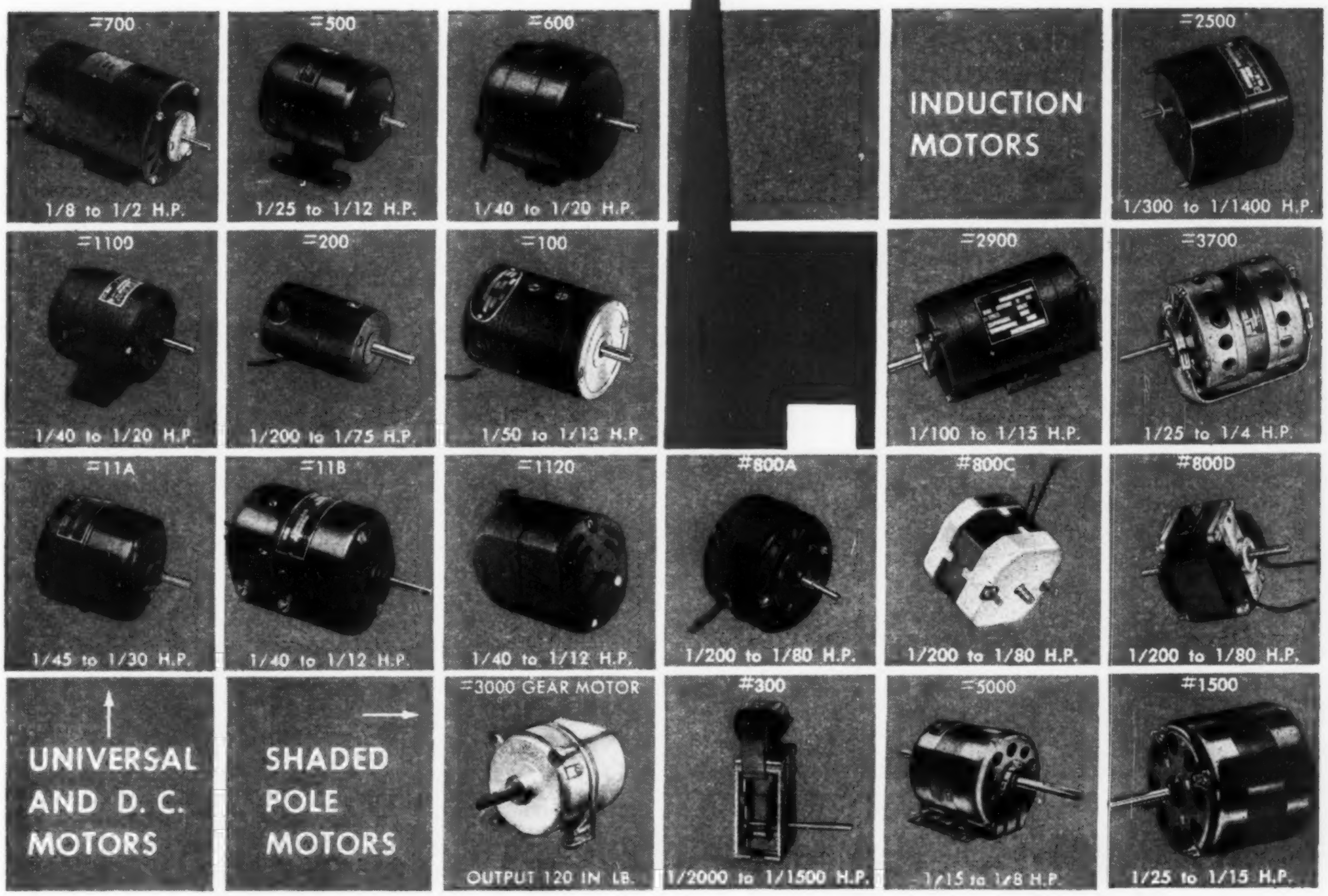

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adjacent to the joint. A Nicrobraz composition with boron, carbon, and silicon removed would melt at about 2550 F. Diffusion during brazing or by subsequent heat treating, simply reduces the concentration of these critical elements and thus raises the "remelt" temperature of the joint.

From a paper entitled "High Temperature Brazing Applications" presented at the SAE National Aeronautic Meeting in Los Angeles, Calif., October, 1953.

Human Factor In Design

By Dickey Dyer
The Work-Factor Co.
and W. H. Compton
Wood-Compton Co.
Cleveland, O.

PROPER arrangement of controls and meters is most essential in order to fully utilize human efficiency in machine operation. Many human factors must be considered in design. Although there is much material available on the design aspects of machines themselves, the subject of the combination of man and his machine is often neglected.

Machine Planning: In the layout of a machine for the best man-machine combination, there are six factors to be considered.

1. All controls and meters should be located within a practical working distance from the operator.
2. Within the practical working

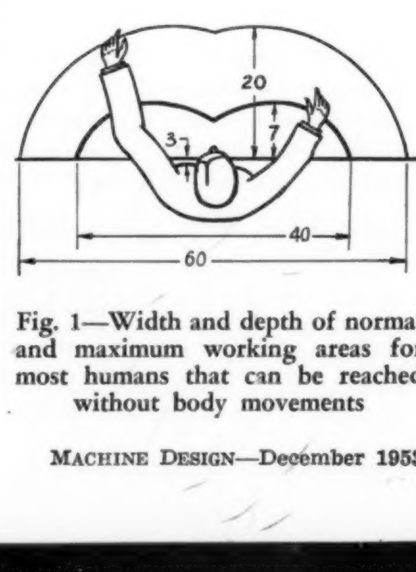


Fig. 1—Width and depth of normal and maximum working areas for most humans that can be reached without body movements

distance, each control and meter should be located in the position where it may be most efficiently used.

3. Where two or more controls or meters have the same optimal position, the most important and most frequently used items should be given the preferred positions.
4. Controls and meters should be grouped in patterns that make for the easiest operation and observation from the point of view of the operator.
5. Other parts of the body should be utilized when the hands are over-loaded.
6. Any possible confusion of controls or meters by the operator should be avoided by proper design or placement.

Human Limitations: The placement of the operator in a position to reach each control properly, to load and unload parts, and properly view the meters at the same time is not simple. For example, this problem exists when considering the maintenance crew who service large electrical or hydraulic control panels for automation lines where a few minutes of down time is a considerable loss.

One method of determining the reach of the human to be considered is to use the data on body sizes collected in various anthropological studies. Normal and maximum working space in all dimensions must be considered. In *Fig.* 1 is a view of the expected reach of most human beings in the horizontal plane. Here the

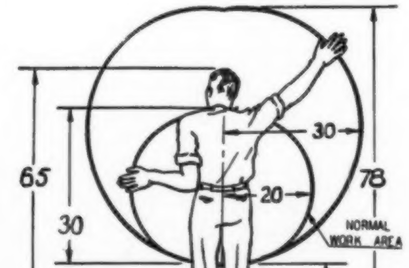
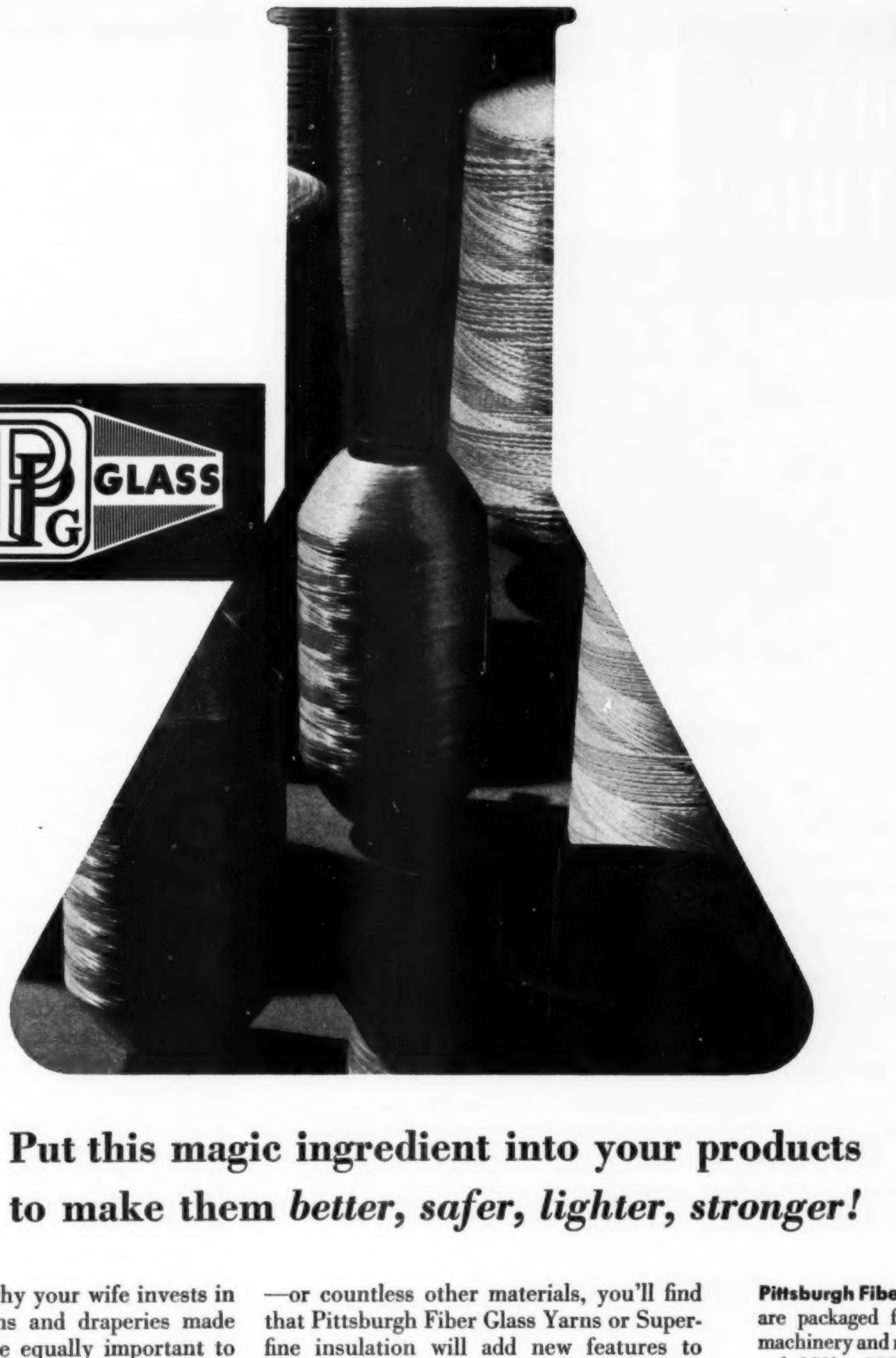


Fig. 2—Width and height of normal and maximum working areas for most humans while standing. Eye level is at 61 inches



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semicircular arcs of the maximum and normal working areas are defined. These areas apply to an operator who is standing in a fixed position without moving his trunk. Dimensions shown are based on minimum instead of average in order to accommodate 95 to 97 per cent of all humans.

Man is more than just hands, fingers, and arms; he also has a trunk, knees, legs, and feet to operate controls. Locations of these are shown in *Fig.* 2, where range of the normal and maximum working areas is shown for the vertical plane.

The location of the knee is about 13 inches from the floor, pivoted at the hip, which is 32 inches high. If he is located on a seat 30 inches high, he can operate in the same range as an individual who is standing on his feet. In this particular case it is well to provide a footrest. When one is sitting, part of the working area is eliminated by the knees.

Human Responses: Although it has been claimed that humans respond faster to audible than visual signals, meters and indication lights are the most common signals on most machines.

In order to determine the best location of visual indicators we must note human seeing ability.

From a paper entitled "Economics of Location of Control Devices on Machine Tools" presented at the Sixth Annual AIEE Conference on Machine Tools in Cleveland, O., October, 1953.

Practical Approach to Component Design

By J. C. Garrett

President
Garrett Corp.
Los Angeles, Calif.

COMPONENTS may be defined generally as highly technical products which are fundamental, functioning parts of the airplane, electronic equipment, engine, or missile. They are designed by the component manufacturer and sold as completed articles to meet cer-

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tain specific problems and conditions.

When Orville Wright made his first powered flight fifty years ago, he used only five components—a stop watch, a veeder counter, and anemometer, a magneto, and a radiator. Fifteen years later components comprised about 10 to 15 per cent of the cost of the finished airplane in World War 1. Today a conservative estimate of the cost of the components in a modern jet fighter would be approximately 50 per cent of the fighter's cost, less engines and radar equipment. If we were to include radar and guidance equipment, the total would easily exceed the cost of the airframe.

Certainly a component manufacturer is out of place in debating the merits of what is required in the fighter, bomber, or missile of today. However, we are convinced that whatever the mission may be, there will be an increasing demand for lighter, more compact and more reliable components to take over where man leaves off.

Standardization is of paramount importance. The present trends are frightening to those deeply concerned with supplying the aircraft industry's needs for specialized components.

Development Costs of Components: At our own AiResearch plant we are building over 850 different models of components for use on airplanes and missiles today. Of this 850 we are in production on over 250 types of electric actuators alone, over 350 different models of air valves and regulators, and over 45 different types of cooling systems. When the equipment produced by competitors is added to our total, the spare parts problem this poses is staggering. The logistic and cost problem is increasingly complicated. Also each new model requires separate development and engineering and separate testing before it goes into production. Our average cost for developing and qualifying a new model of hot air valve today—being required to operate at around 1000 F —is approximately $30,000 to $40,-000. It cost us over $100,000 to develop and qualify a new cooling

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turbine. After careful consideration and due allowance for the necessary complexity of the modern weapon, a conservative estimate of possible reduction in the number of types would be one-third. Hundreds of millions of dollars and vital production man-hours thus could be saved in an all-out emergency.

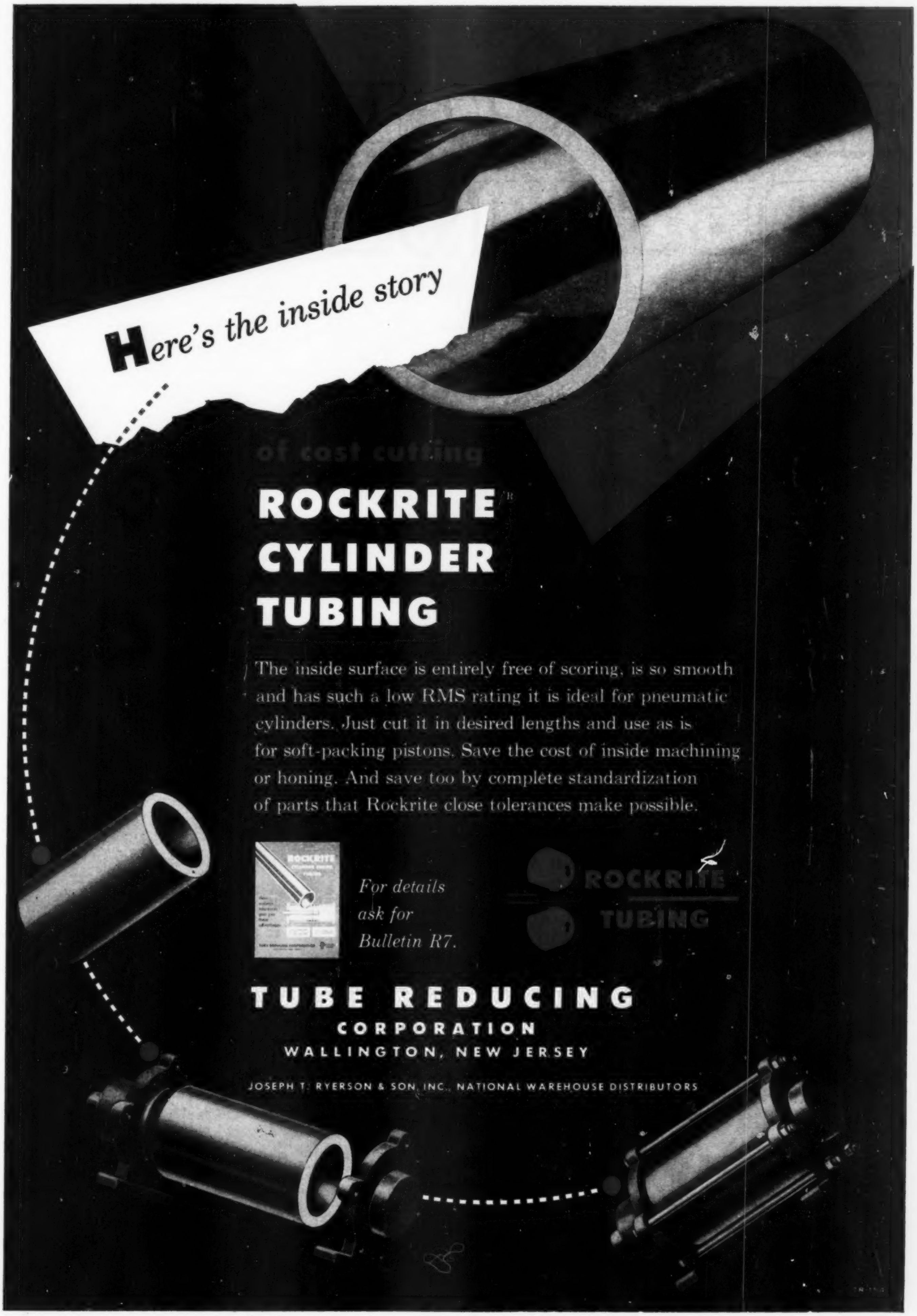
Advances in Design: Great strides have been made within the past few years in the miniaturization, operating effeciency, and durability of many components. Efficiency of fans used in aircraft cooling systems alone has increased from 30 to 90 per cent in the past seven years. The cabin pressure regulator of today does that same job under much more severe operating conditions at a weight of only 1 pound. It operates with three times the accuracy, and costs ¼ what the World War II regulator cost. Refrigeration systems have increased 700 per cent in capacity per pound of weight in a few hectic years. Ram air turbines for emergency power supply for jet aircraft control systems weigh only 1/10 as much as the emergency systems of four years ago. Certainly we can look forward to the continuation of these healthy trends.

From an address presented before the Second Annual Air Force Association Airpower Symposium, Washington, D. C., August 1953.

Selecting Metals for High Temperatures

By E. H. LaBombard
Assistant Supervisor
Structures Section
Douglas Aircraft Co. Inc.
Santa Monica, Calif.

THE MOST apparent effect of high operating temperatures on structural materials is the reduction of static properties. Curves in *Fig.* 1 compare the change in static properties of titanium with other common materials. In *Fig.* 2, the variation in modulus of elasticity is shown for these materials. The drop in modulus for all these ma-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terials is in the order of 25 per cent.

Strength-Weight Properties: The result of reduced strength properties is a consequent increase in structure weight. A strength-weight ratio is not too useful as a parameter but calculations employ-

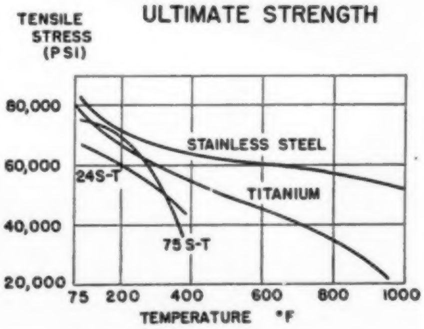

Fig. 1—Short-time ultimate tensile strength properties of 302 stainless steel, RC-70 titanium, 24STAL and 75STAL

ing it are sometimes significant. In this case, they indicate that by selecting the best material from strength-weight consideration corresponding to a particular operating temperature, the structure can be expected to weigh from 10 to 100 per cent more than a compar-

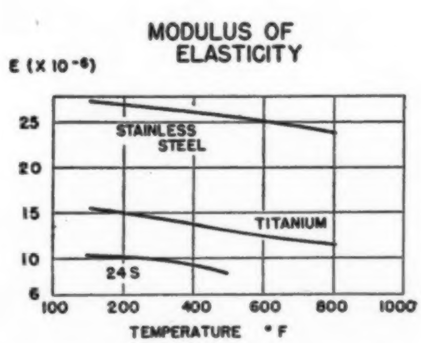

Fig. 2—Young's modulus for three metals based on limited data

able low temperature structure. It is interesting to note from *Fig.* 3 that because the change in strength with temperature is not linear, the best material may be aluminum, steel, or titanium depending on the particular operating temperature involved. These curves indicate

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that commercially pure titanium has a weight advantage in the temperature range of 450 to 650 F. This observation is based on ultimate tensile strength. On the basis of yield strength the range is extended somewhat. However, this also shows that titanium is not the material to solve all high temperature structural problems. Even the presently unavailable titanium alloys are not as good as stainless

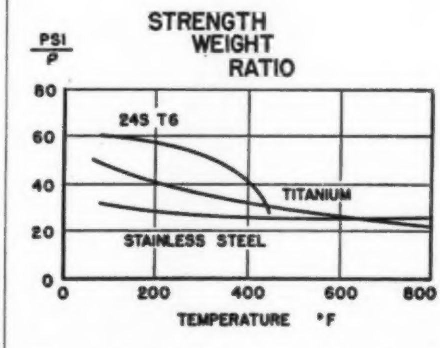

Fig. 3—Strength-weight ratios based on curves from *Fig.* 2 and density

steel at the higher temperatures. When a strictly gage-for-gage substitution of titanium for steel in low-stress applications is made, large weight savings are possible.

Corrosion Resistance: Probably more significant, however, than the reduction in static properties with temperature increase is the effect of time at temperature on the materials. Two significant properties that are functions of time at temperature are creep and corrosion resistance. Because aircraft structures are designed to criteria of permanent deformation, the effect of creep becomes a significant if not a dominating factor.

Corrosion resistance is another property affected by continuous exposure to high temperatures. In general the resistance to corrosion is reduced, and the corrosive environment is much worse. Most of our corrosion protection methods are unsatisfactory and service problems can be expected to multiply.

Reduction in corrosion resistance at higher temperatures is due to two things. First, of course, the elevated temperatures increase the rate of corrosive reactions. Sec-

ond, and perhaps more significant, after time exposure to elevated temperatures most materials undergo a metallurgical change that reduces corrosion resistance. The critical conditions at which this change occurs vary with temperature and time of exposure for all materials. For this reason, the best material for corrosion resistance should be selected on the basis of actual operating temperature.

The protection of materials against corrosion offers quite a problem. There are few paints that can last in corrosive environments. Cadmium plate as a protection for steel, long a standby in the industry, ceases to be useful above 500 F, so other methods must be employed. Metal-sprayed aluminum seems to offer some promise. Corrosion resistance of titanium is excellent; superior under many conditions to stainless steel. Therefore if strength at temperature is not of prime importance, this reason alone can dictate its use in many places.

From a paper entitled "The Effect of Temperature on Aircraft Structural Design" presented at SAE National Aeronautical Meeting in Los Angeles, Calif., October, 1953.

Applying Strain Gages In Design

By Ira H. Abbott

National Advisory Committee for Aeronautics

APPLICATION of strain gages is usually associated with direct measurement of strain or stress in structures. However, recent developments have brought forth some high-temperature applications which offer interesting possibilities. In addition, experience has shown that strain gages can be effectively used to measure dynamic loads.

High-Temperature Applications: Measurement of dynamic strains at temperatures up to approximately 1500 F has been a subject of considerable interest during the development of the modern turbojet en-

Design Abstracts

gine. The need for an instrument of this type was evident when it was realized that many of the turbojet failures were the result of destructive vibration of the turbine blades.

Development of the high-temperature strain gage proceeded along somewhat the same lines as the development of the low-temperature, resistance-wire strain gage. It was necessary to evaluate many cementing materials, wires, and fabrication techniques before a satisfactory gage was developed. The high-temperature strain gage in its present form is comprised of 0.001-inch diameter Karma strain sensitive wire, 0.010-inch diameter Karma lead wires, National Bureau of Standards L6AC precoat ceramic and Quigley No. 1925 cover ceramic. The complete gage consists only of a strain-sensitive, wire grid embedded in ceramic on the test specimen. There is no winding form or extraneous material within the gage which might deteriorate and cause malfunction.

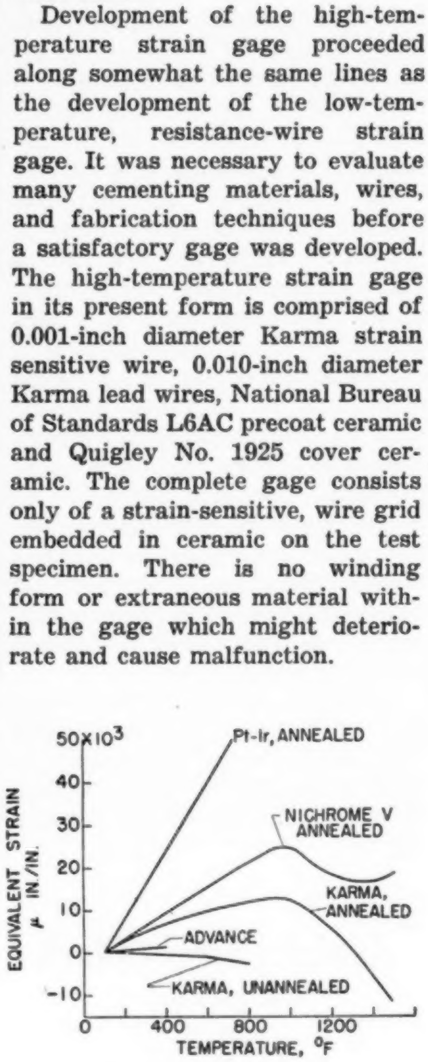

Fig. 1—Resistance change curves for common strain gage wires

This gage has been found to perform satisfactorily when mounted on the turbine blades of an operating engine, where it must withstand considerable punishment and still furnish reliable information. First, the gage must adhere to the blade despite the high centrifugal force field of up to 50,000 g's. Second, the gage must operate at temperatures up to 1500 F and retain satisfactory mechanical strength. Third, the gage must withstand the extreme erosion forces of the hot gas stream, and finally, it must withstand severe

thermal shocks during starting and shutdown operations.

The wire alloy Karma in the "as received" or unannealed condition has a low temperature coefficient of resistance, *Fig.* 1, when compared with Nichrome V or Pt-Ir. When used only up to 800 F, the temperature coefficient is comparable to that of Advance, providing the possibility for static strain work in that range. The resistance change of the gage with temperature is sufficiently small to permit adequate corrections or compensations in certain types of static work. In addition, the wire has a very high specific resistivity (approximately 800 ohm/cir.mil.ft.) and excellent corrosion resistance to the cements used.

The method of preparing the strain sensitive grid for mounting is shown in *Fig.* 2. Two small rods are fastened in place behind the frame, the wire is initially wound around these rods and the lead wires are placed in their respective holders. Fine strands of silk are then used to tie back each loop to the frame and the rods are removed. The loops are tightened by pulling the silk strands, and the gage is then ready for mounting.

A typical mount on a turbine blade is depicted in *Fig.* 3, showing, in particular, the method of handling the lead wires. One of the leads is resistance welded to the blade, while the other is welded to a secondary lead passed out through a ceramic lined inconel conduit to a terminal plate in the center of the turbine wheel.

Dynamic Loads: The need for measurement of dynamic loads led to a desire for a less cumbersome method than pressure distribution. The wire-resistance strain gage, when properly installed and calibrated, offers a useful tool for structural load measurements which can, in turn, be converted to dynamic loads, provided the structural weight distribution is known and the acceleration distribution has been measured. Although the application of strain gages for load measurements is similar to their use in stress determination, a somewhat different approach is required. In stress measurement, a

single strain gage is frequently employed to measure the stress in a member. In load measurements, four active-arm bridges are generally applied on the principal structural members and temperature compensated in order to obtain high sensitivity and relative freedom from the effects of uniform structural temperature changes.

Strain Gage Location

The first step is the selection of the gage location, which depends on the measurements to be made. It is necessary to locate the gages at positions where the stress levels will be adequate to obtain good sensitivity and, at the same time, away from areas of local stress concentrations. Gages should not be installed near fittings or fasteners that may develop clearances and consequently change the stress distribution. Ideally, it would be desirable to place the gages at a position such that a shear bridge would respond only to shear, a moment bridge only to moment, and so forth, but such locations cannot generally be found in complex structures.

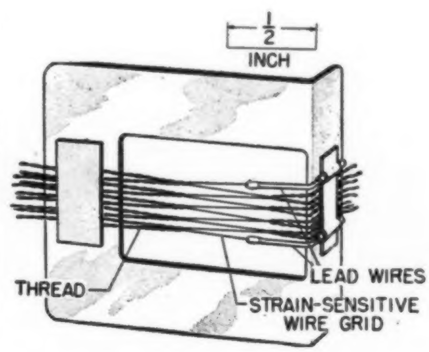

Fig. 2 — High-temperature strain gage prepared for mounting

Next, through calibration procedures, a system of empirical equations must be developed for determination of load and load distribution from the gage readings. It is apparent that enough bridges must be installed to permit the development of appropriate equations relating load and bridge response.

In step three the application of the calibration loads must be con-

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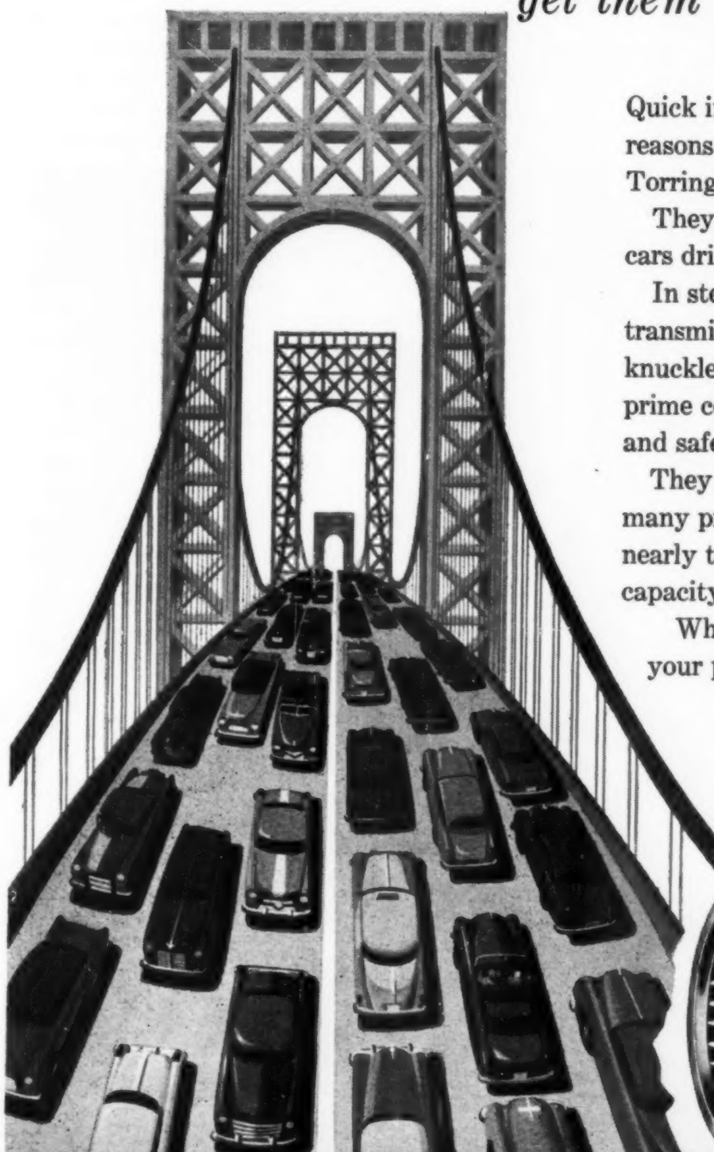

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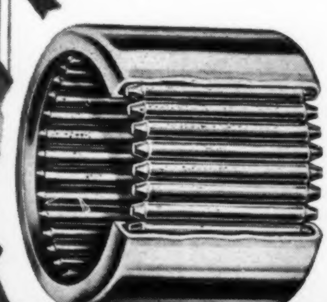

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Design Abstracts

sidered. These are usually applied with jacks through pads large enough to prevent local buckling. In order to assess any possible effect of elastic lag, application of these loads by increments is recommended. To provide data for evaluating the effects of carry-over, the loads should be applied independently to each side of the structure and then to both sides simultaneously.

The next step involves evaluation of the preliminary calibration data. Plots of the ratio of the bridge responses to load against distance to the point of load application along any straight line on the structure (termed influence coefficient plots) are useful guides to the characteristics of the bridges and assist in determining the form of the calibration equations.

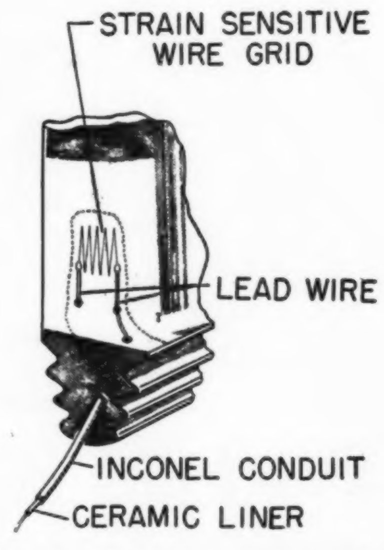

Fig. 3—Turbine blade mounting technique for high-temperature strain gage

The final steps in the calibration procedure depend upon the results of the preliminary calibration in relation to the electrical recording equipment available and the number of different loads which it is desired to measure dynamically. It may be desirable to record each bridge output independently, and to calculate the desired loads from the calibration equations, using automatic calculating machines. It may, alternately, be desirable to combine bridges, using duplicate

bridges where necessary, into calculating electrical networks that permit the desired loads to be recorded directly. Or, a combination of the two methods may be used.

Direct current systems are used exclusively for flight aerodynamic loads measurements by the NACA because they provide the most stable circuit characteristics. Provision must be made for the possibility of changes of sensitivity or of zero drift in the recording apparatus. Changes of sensitivity may result from changes in supply voltage to the strain gage bridge and to the recording galvanometer elements. Drift may result from temperature effects on the galvanometer element and from temperature effects on the structure.

Accurate results are obtainable only through the exercise of continual vigilance and attention to detail. Perhaps the greatest danger is that some important factor or effect that will cause errors in the results may be over-looked. These errors may not be recognized if the particular calibration procedure selected is not sufficiently inclusive to detect them. It must also be remembered that under dynamic load conditions the strain gage system will sense only the loads which the structure will follow. For this reason the method is limited in response to the basic frequencies of the structure. Also, the inertia loads, while theoretically correctible, may be difficult to extract under dynamic conditions because of the requirements for many accelerometers and the tedious determination of polar moments of inertia. Nevertheless, the method works very well if its limitations are understood.

From Advisory Group for Aeronautical Research and Development Memorandum, "Some Application on Strain Gages in Aeronautical Research."

Correction

In the abstract entitled "Designing Scales and Charts," November, Page 264, *Fig.* 2 and its caption were not in agreement. In the figure *a* and *b* should be interchanged.

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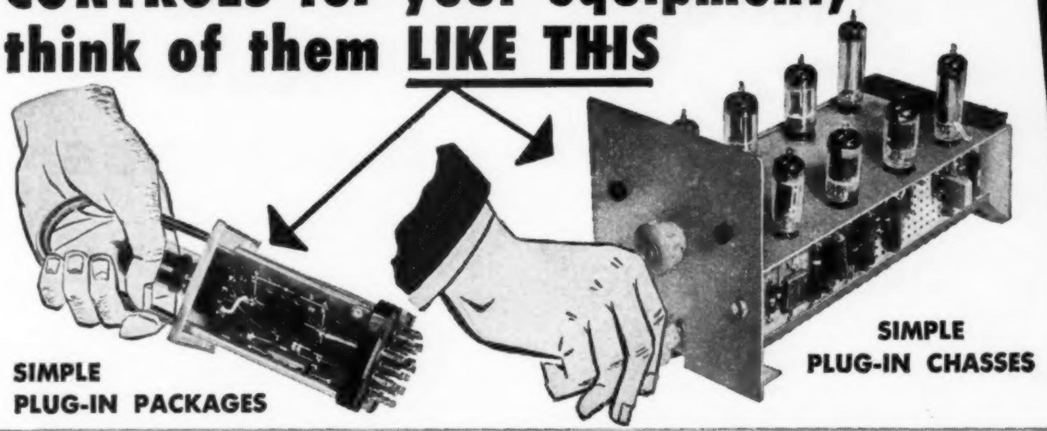

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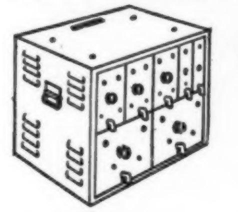

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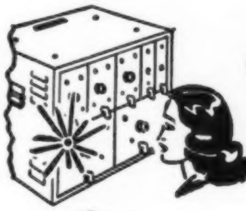

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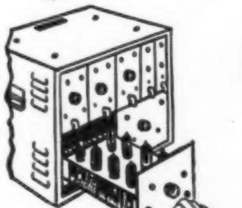

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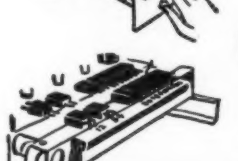

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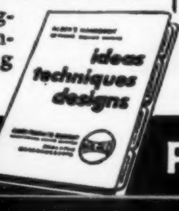

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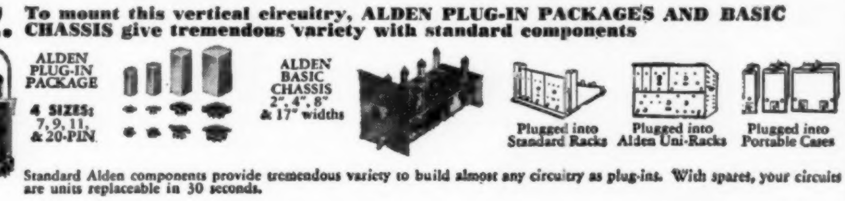

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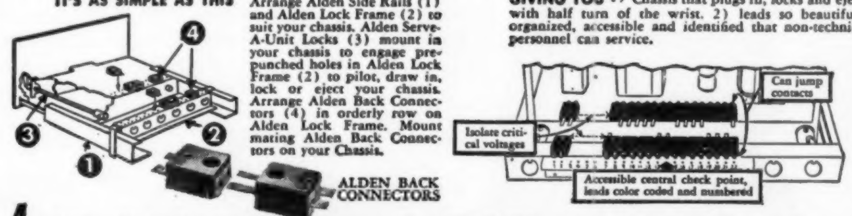

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Mower-Leaf Mulcher: Power-propelled 20-in. Whirlwind rotary blade mower cuts and trims grass, and mulches leaves. Has a 2½-hp, 4-cycle engine with a recoil starter. Leaf mulcher attachment pulverizes 30 or more bushels of leaves a minute, pulling the leaves under housing, suspending them and chopping them into a fertilizing mulch. *Toro Mfg. Corp., Minneapolis, Minn.*

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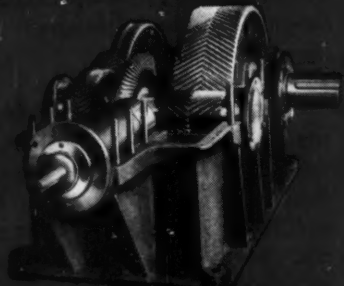
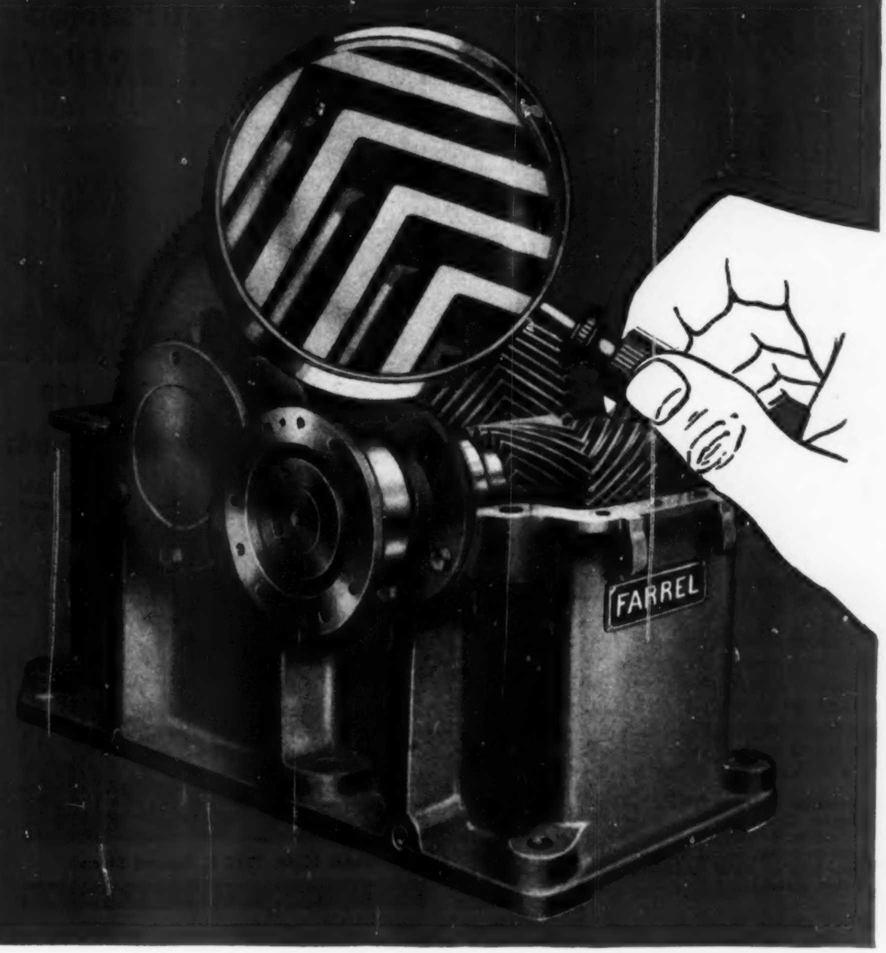
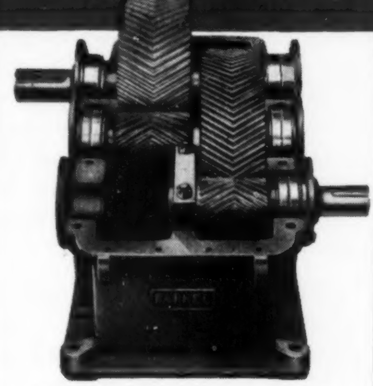
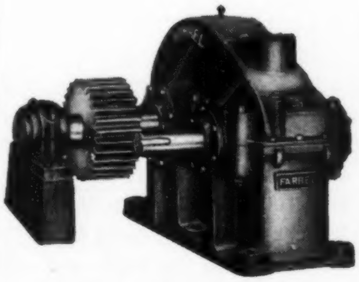


Horizontal right angle unit





Standard double reduction unit



Unit with pinion on extended shaft supported by outboard bearing

Close-up of a way of life for Speed Reducers

Long, trouble-free speed reducer life is vitally dependent on precision generation of the gears.

The gears of Farrel® speed reducers are made by the famous Farrel-Sykes method — a process that assures accuracy of tooth spacing, profile and helix angle. The herringbone design provides evenly distributed pressure over each tooth, from tip to working depth line. This means that there is no tendency for the teeth to wear unevenly and thus shorten the life of the gears.

Shafts and bearings are factored to safeguard against interruption of vital processes. Gear cases are proportioned to withstand repeated heavy peak loads. Joints are sealed to prevent entrance of dirt.

Farrel speed reducers are made in a number of types, with a wide range of ratios and capacities. Designs include single, double and multiple reduction units, speed change units having two or more selective speeds, right angle drives, and drives to meet special requirements.

Wherever power transmission must be smooth, quiet, and vibrationless under all conditions of load and speed, *specify Farrel speed reducers.* For further details, send for a copy of bulletin 449.

FARREL-BIRMINGHAM COMPANY, INC., ANSONIA, CONN.

Plants: Ansonia and Derby, Conn., Buffalo, N. Y.
Sales Offices: Ansonia, Buffalo, New York, Boston, Akron, Detroit, Chicago, Memphis, Minneapolis, Portland (Oregon), Los Angeles, Salt Lake City, Tulsa, Houston, New Orleans

FB-840

Farrel-Birmingham®

for varying weather conditions. Plastic air delivery grilles cause air to move up or down as desired, and vanes direct air in any horizontal direction. Centrifugal blower type fan is housed in a section behind the cooling coil and surrounded by sound-absorbing insulation. Return air enters the unit along the bottom front edge of the cabinet, and stale air is exhausted through a small panel on the cabinet top. Twin series units are available for 115, 208 and 230-v operation; Super series units are available for 115-v operation only. *Frigidaire Div., General Motors Corp., Dayton, O.*

Portable Heater-Ventilator: Palmaire Jr. heats, humidifies, circulates air and can be used as a spot evaporative cooler. As a heating unit it has a 1350-w heating element that produces 4600 Btu. As a humidifier it circulates 2 gal of water from its reservoir over a specially developed spun glass pad. Air is filtered as it is drawn through the pad for distribution by the fan. This humidified air can be warmed by means of the heating element. Unheated, the unit acts as an evaporative cooler. Used as an electric fan, it circulates 700 cfm of cool, filtered air. The unit weighs 28 lb. *McCray Refrigerator Co. Inc., Kendallville, Ind.*

Room Air Conditioners: Redesigned models in 1/3, 1/2, 3/4 and 1-hp sizes provide increased cooling and dehumidification, as well as a lower velocity air flow, resulting in quieter operation. Three circular louvers can be rotated independently to direct air flow in various patterns. Five ventilating conditions may be selected: maximum cooling, normal cooling with ventilation, cooling with extra dehumidification, fresh air ventilation, and stale air exhaust. Preselected temperature is maintained by thermostatic control. All models are 14 7/8 in. high and 25 5/8 in. wide. The three smaller units are 31 7/8 in. in overall depth and extend 13 3/4 in. into the room from the window line; the 1-hp model is 32 7/8 in. in overall depth and extends 14 3/4 in. into the room. The four models are designed for

New Machines

use in rooms ranging in area from 200 to 700 sq ft. The ¾-hp size will also be available with reverse cycle operation for heating. *Major Appliance Div., General Electric Co., Louisville, Ky.*

Materials Handling

Electric Skid Truck: Rider type truck transports semi-live skids in congested areas and narrow hand truck aisles. A toggle lever, mounted on the front of the tractor unit, engages the wheelless end of the skid and raises the legs off the floor for easy movement. Truck handles loads up to 4000 lb and can turn in a 40 x 48 in. intersecting aisle with a skid 24 in. wide x 60 in. long. Travels at speeds up to 5 mph. *Raymond Corp., Greene, N. Y.*

Revolving Crane Truck: Capacity is 2000 lb; overall height is 68 in. Truck's hook, which is 60 in. high when raised and 10 in. high when lowered, projects 24 in. over either side of the truck and is raised and lowered by a pump and motor which pumps fluid into a double-acting ram inside the main mast. The hook does not depend upon gravity for lowering. Boom has a complete angular travel of more than 180 degrees. Truck body is 36 in. wide; 92¼ in. long overall and has a 57-in. long platform. It weighs 4800 lb including battery. Loaded, speed is about 2½ to 3 mph. *Market Forge Co., Everett, Mass.*

Metalworking

Milling Machine: Keller type BG-21 is used either as an automatic electric tracer-controlled unit or as a regular milling machine. Tracer milling is accomplished by either following a sheet metal template or side walls of a model with a profiling tracer and cutting the duplicate shape with the side of an end mill, or by using a three-dimensional tracer to follow a full model in a series of parallel passes with the spacing preset by the operator. Horizontal construction provides maximum support for spindle head. Movable weight is constant, since workhold-

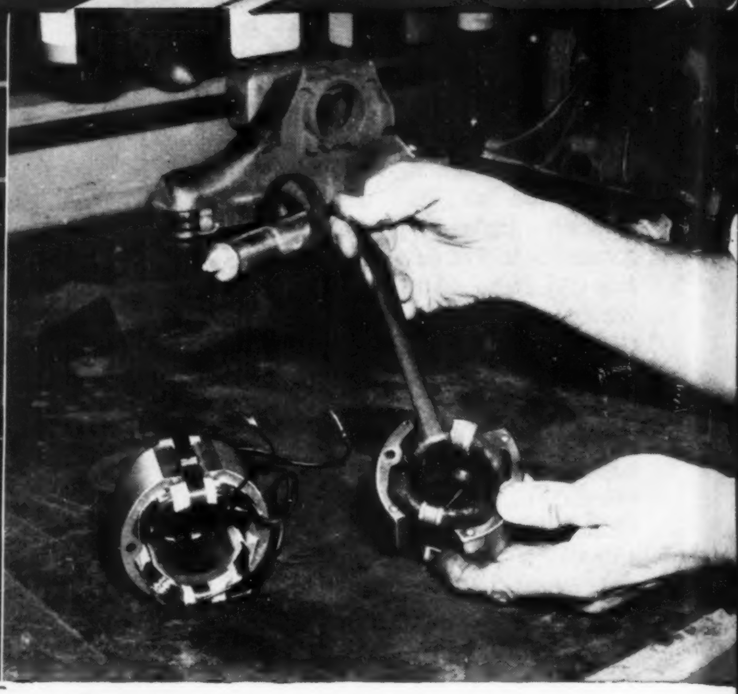
New Machines

ing table is stationary and the spindle head carries the cutter along the work. Work and model need not be located in exact relation to each other, inasmuch as both the spindle and tracer are adjustable. Machine has a central oiling system. All sliding surfaces have phenolic-to-metal contacts, and all lead screws run through molded phenolic nuts. Available in five sizes ranging from 4 x 2½ ft to 10 by 4 ft. *Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford, Conn.*

Horizontal Spindle Grinder: Head slide has hydraulic cylinder to hold head against a micrometer adjusted stop which controls amount of stock to be removed. Grinding head consists of a heavy spindle, 3.937 in. in diameter at the wheel collar end; a double roller type bearing for radial load; and two adjustable thrust ball bearings. Hydraulically operated work table travels on dovetailed ways and has a working surface approximately 15 in. wide x 18 in. long with T-slots for mounting fixtures. Grinding head is driven by a 25-hp motor. Stock removal is 3/64 to 1/16-in. maximum overall; accuracy is 0.0003 to 0.001-in. for flatness; 0.010-in. for uniformity. *Gardner Machine Co., Beloit, Wis.*

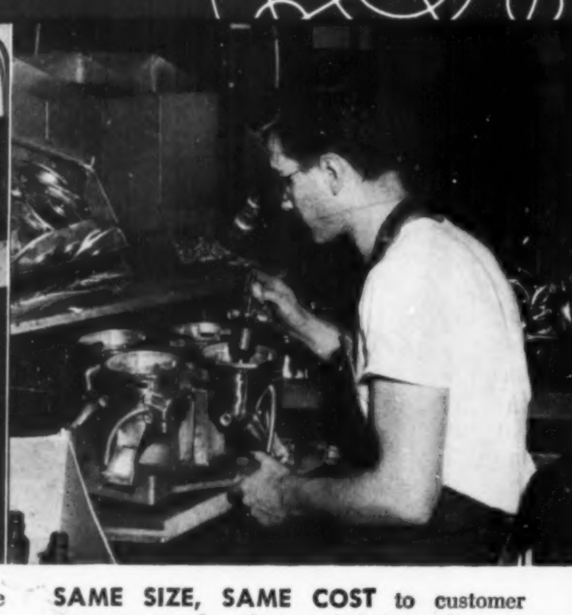
Straightening Machine: For squares, flats and structural shapes such as angles, channels, T-sections, etc. Straightens squares from ½ to 2½ in. and flats up to 4 x 1 in. at speeds up to 400 fpm. Outboard bearings facilitate roll change while retaining machine strength approaching that of a unit with a double housing. Roll change necessitates only the loosening of a single bolt for each bearing. When each is removed, the shaft is fully supported by the two main bearings. Pass line consists of four bottom driven rolls with five top pressure or deflecting idler rolls. Pressure adjustments are controlled by handwheels. Top roll housings are adjustable longitudinally on the base to provide for sharper deflection of small sections. Top rolls have in-and-out adjustment along the center line of the roll shaft so that they may

(*Continued on Page* 406)



SCOTT AND FETZER PRESENTS PROBLEM to G-E Engineer: how to get a more powerful, yet cooler-operating motor for the Scott and Fetzer Kirby Home Sanitation System in same frame size.

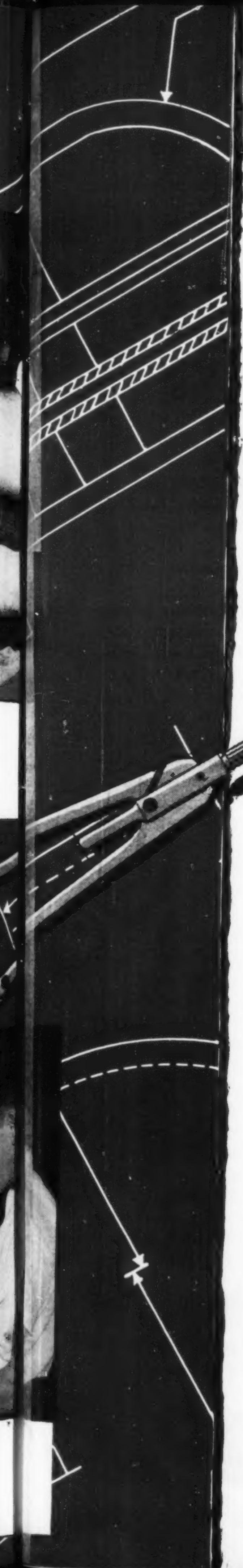
G-E ENGINEERS REDESIGNED the motor windings, and eliminated the tape from the field. This solution was worked out by the G-E Specialty Motor Department in Fort Wayne.



12% MORE POWER was achieved by the Specialty Motor Department's design, yet the temperature rise was cut by almost 50%.

SAME SIZE, SAME COST to customer meant no costly redesign of their product, or extra cost for its better motorization.

INTERCHANGEABLE PARTS, in new motor, eliminate any service problem with older units now in the field.



(Continued from Page 403)

be staggered in relation to the bottom rolls and to each other. *Sutton Engineering Co., Bellefonte, Pa.*

Cutoff Machine: Model M-35 has direct-drive 3½-hp geared-in-head motor which delivers full power to the cutting edge. Machine uses a 12-in. abrasive wheel for cutting ferrous metals and a 12-in. steel saw blade for cutting nonferrous metals. Capacity in ferrous metals is 1½-in. solids, 2½-in. OD pipe and 2½-in. structural shapes. Capacity in nonferrous metals is 2½-in. solids and 3-in. tubing and extrusions. *Stone Machinery Co. Inc., Manlius, N. Y.*

Plant Equipment

Slab Breaker: Pneumatic impact machine has span of approximately 20 ft and can be built with various stroke lengths ranging from 42 to 84 in. depending on impact required. Delivers 50 to 60 impacts per minute with a 64,000 ft-lb impact. In operation, crane picks up slab and places it on anvil. Machine then moves into position, breaks slab to desired size, and withdraws. Hydraulic bulldozer is used to push broken material off slab onto cars. *R.P.B. Corp., Los Angeles, Calif.*

Motor Driven Pumps: The 3600 Series MAHBRV No. 5 and No. 6 pump units have a range of capacities from 70 to 226 gpm at 60 psi. Each size has three interchangeable gear ratios. Features include totally enclosed gear reduction running in oil; antifriction bearings throughout gear reduction; indoor or outdoor operation without need of pump house; sturdy bedplate for rigid mounting of both pump and motor; compact design. Pump head has two equal size pumping gears which operate in axial hydraulic balance and are keyed to the shaft with sliding fit, facilitating inspection or replacement. Pump head is available with or without built-in relief valve. *Geo. D. Roper Corp., Rockford, Ill.*

Processing

Vacuum Furnace: Airfre high-temperature electric furnace pro-

duces temperatures up to 2500 F and vacuums up to 29 in. Higher vacuum can be supplied for special applications. Furnace and vacuum system can be used independently of each other. Available in various sizes, complete with vacuum pump and control equipment such as gages, automatic pressure controller, Variac, ammeter, timer and automatic temperature controller. *K. H. Huppert Co., Chicago, Ill.*

Steam Cleaner: Self contained Model LH discharges up to 150 gph of balanced cleaning solution. Cleaning pressures range from 30 to 100 psi for applications from paint stripping to rapid cleaning of heavily encrusted or grease coated parts and surfaces. Heating coil, wound in a helical pattern, provides thermal efficiency of 75 per cent minimum. Full working pressure is attained in two minutes from a cold start. Models are available oil and gas-fired; electrically and gasoline engine driven; in portable, stationary or trailer types. Each can be equipped with remote nozzle control. Thermostat controls prevent overheating by cutting off fuel supply to burner in the event of excessive pressure or temperatures due to water supply failure. *Clayton Mfg. Co., El Monte, Calif.*

Mold Forming Machine: Spomatic unit automatically mass produces uniform standard molds. It makes accurate copes and drags simultaneously, sets the cores, closes the molds and puts them on the pouring conveyor. After the molds have been poured and put through a cooling cycle, it knocks out, cleans and separates the flasks and feeds them back into their respective ramming machines for immediate re-use. *SPO Inc., Cleveland, O.*

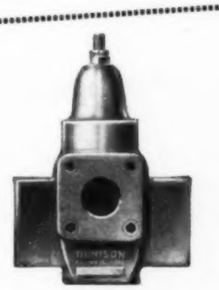
Heat-Treating Furnace: Model FG-430 handles all work including high-speed steels, at continuous heavy duty up to 2500 F and can handle higher temperature work over shorter periods. Gas-tight ports connect atmosphere reducing sources. Counterbalanced door is foot controlled and pneumatically operated. Flame curtain is located immediately below the opening and ignites automatically when the door

For lasting efficiency with less maintenance, specify

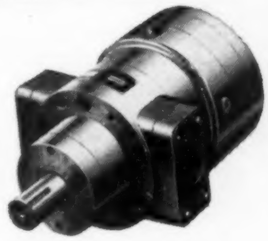
DENISON HydrOILic PUMPS, MOTORS, CONTROLS

For circuit pressures to 5000 psi

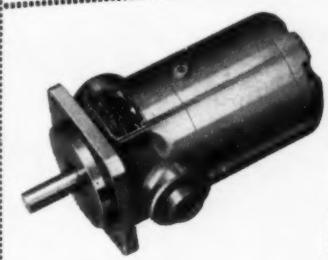
SURGE-DAMPING CONTROLS
Industrial and Aircraft Types
Eliminate Destructive Hydraulic Pressure Shock
1/4" to 1" Sizes
For Pressures to 5000 psi
Catalog VS-1-A



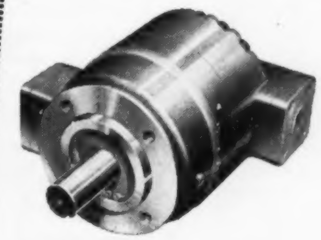
Direct-Operating
PRESSURE CONTROLS
1/2" to 1 1/2" Sizes
For Pressures to 2000 psi
Catalogs VRD-1 through VRD-8



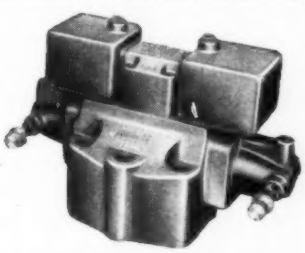
High-Pressure Axial-Piston
PUMPS
Constant or Variable Volume
For 2500, 3000 and 5000 psi Circuit Needs
3.5 to 35 gpm Capacities
Catalogs P-4 through P-4-10



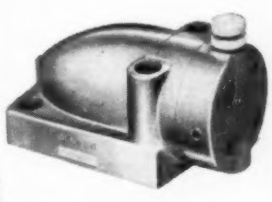
High-Torque Axial Piston
FLUID MOTORS
Torque Ratings 13.8 to 112 inch-pounds per 100 psi (about 3 to 180 hp maximum)
For Pressures to 5000 psi
1500 to 3000 rpm
Catalogs FM-1, FM-2, FM-3-A



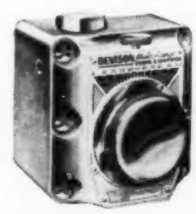
Dual-Purpose, Vane-Type
PUMP/MOTOR
Ready for either pump or motor use without change
Shaft rotation in either direction
Four Sizes, 11 Models
Pumping Capacities: 3 to 82 gpm.
Motor Torque: 13 to 257 inch-pounds per 100 psi. Pressures to 2000 psi.
Bulletin P-5



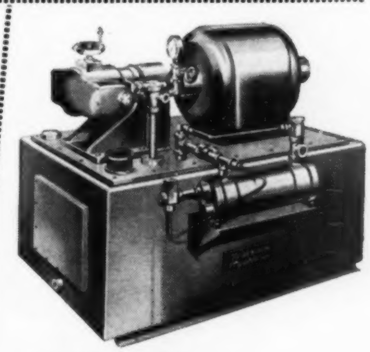
DIRECTIONAL CONTROLS
Manual, Mechanical, Electric or Pilot Control. 1/4" to 2" Sizes
For Pressures to 5000 psi
Catalog VD-3



PRESSURE REDUCING VALVES
3/4" to 1 1/2" Sizes
20 gpm to 65 gpm Capacities
For Pressures to 5000 psi
Catalog VPR-1



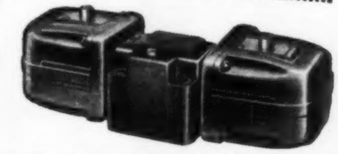
Multi-Range
FLOW CONTROLS
Full-Scale Regulation—Adjustable for Varying Input
1/4" to 3/4" Sizes
For Pressures to 3000 psi
Catalogs VFC-1, VFC-2



PUMPING UNITS
Constant and Variable Volume
46 Models—2 to 35 gpm Capacities
For Pressures to 5000 psi
Bulletins PU-3-A, PU-4 and PU-5

PROVED IN USE

under industry's toughest operating conditions



Pilot-Operated, Solenoid-Controlled
DIRECTIONAL VALVES
Eliminate Spool-Sticking
3/4" to 1 1/2" Sizes, For Pressures to 5000 psi
Catalogs VD-1-1 to VD-4-2

Denison HydrOILic Pumps, Motors and Controls are noted for their combination of high efficiency and rugged dependability.

In addition to fine performance under continuous, heavy-duty use, Denison HydrOILic Equipment offers space-saving design that simplifies circuit layout, makes installation easier, and minimizes maintenance problems.

Because so many users verify the extra value built into Denison Pumps, Motors and Controls, we continue the challenge made so often before—*let comparison prove to you* why they are first choice for so many needs of every type. Write today for full details on Denison HydrOILic Equipment to meet your requirements.

DENISON HydrOILics
PUMPS | MOTORS | CONTROLS | PRESSES

Hydraulically Balanced
PRESSURE CONTROLS
3/4" to 2" Sizes
For Pressures to 5000 psi
Catalogs VR-2-B and VR-S-2

"The Finest Money Can Buy!"

The DENISON Engineering Company, 1156 Dublin Road, Columbus 16, Ohio

New Machines

opens. Heating elements, located beneath the hearth and above the chamber, extend completely through the furnace and may be replaced without unloading or otherwise disturbing the furnace. Loading area is 14 in. wide, 24 in. deep and 12 in. high; outside dimensions are 50 in. wide, 52 in. deep and 84 in. high, excluding panels. *Pereny Equipment Co., Columbus, O.*

Testing and Inspection

Gear Tester: S & F gear tester for inspection of all types of fine pitch gears, including spur, bevel and helical; worms and worm gears; flat and round racks; pinion shafts; finished and semifinished gear assemblies; and small gears and pinions used in instruments and dial gages. Test gear is rotated in contact with a master gear, both flanks contacting the master, so that the gears mesh without backlash. Available for either hand or motor-driven rotation of gears. Instrument consists of a bed, a slide which carries the test gear, and a measuring slide carrier for measuring slide and master gear. Interchangeable bed slides are available. Measuring slide travels on ball bearings with pressure maintained by a flat spring in the measuring slide carrier. Total travel of the measuring slide is $\frac{1}{8}$-in. Slide can be clamped at any point within the limits of its travel. Measuring slide is adjustably spring loaded, with pressures ranging from 0 to 8 oz in either direction. *Kurt Orban Co. Inc., New York, N. Y.*

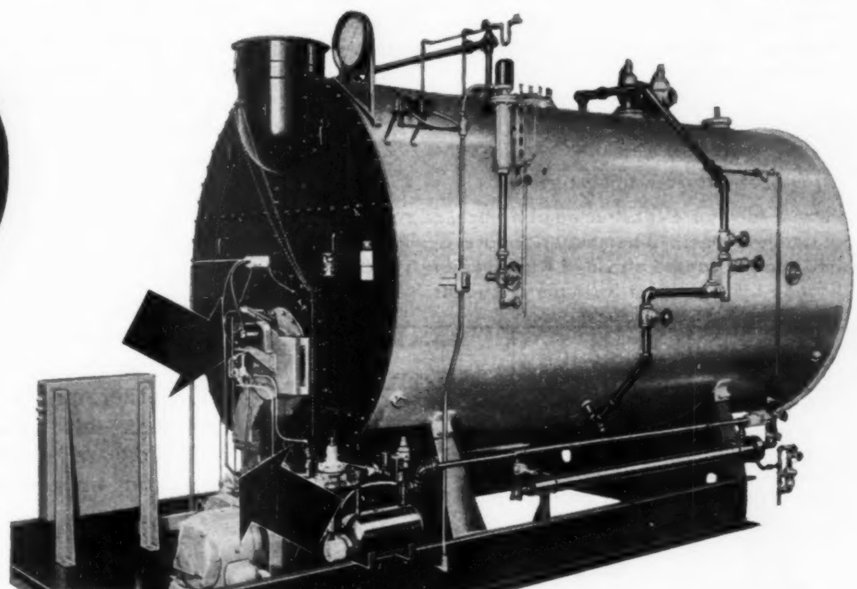
Test Cabinet: Has blower and time-pattern controllers. Temperature range is from 0 to 300 F; humidity range is limited within temperature range and both are controlled to ± 1 F. Temperature and humidity change and length of cycle at a predetermined plotting over 24 to 48-hour periods are automatically cam controlled. Pen recorder automatically graphs actual temperature and humidity. Has two $17\frac{1}{2}$ x $26\frac{1}{2}$-in. deep and two $17\frac{1}{2}$ x 18-in. deep shelves. *Anetsberger Brothers Inc., Northbrook, Ill.*

never a failure with . . .

HEIM *Unibal* ROD ENDS

Heim Unibal Rod Ends are used in the linkage between the blower damper and the oil metering valve, and serve to proportion the air and oil supply for high and low fire operation of this Cleaver-Brooks Self-contained Boiler.

Cleaver-Brooks Company's engineering department is well satisfied with the performance of the Heim Rod Ends, and their service department has never encountered a rod end failure in field applications.



The Heim Unibal Rod End has a single, hardened ball which oscillates freely in all directions in its bronze socket. It corrects misalignment in all directions; it reduces friction and lost motion; and it carries heavier loads.

Your Engineering Dept. will be interested in the Heim Unibal. We will be happy to send you a sample bearing with our complete catalog.

THE HEIM COMPANY

FAIRFIELD, CONNECTICUT . . . . .

MEN OF MACHINES

Charles R. Sutherland

Formerly manager of large motor engineering, **Charles R. Sutherland** has been named assistant chief engineer of Reliance Electric and Engineering Co., Cleveland. Mr. Sutherland is a graduate in mechanical engineering of Fenn College, from which school he also holds a degree in industrial engineering. He received his master's degree in mechanical engineering from Case Institute of Technology. In 1939, following work in electric motor design at Apex Electric and Engineering Co., Mr. Sutherland joined Reliance. He advanced from assistant chief draftsman through intermediate engineering assignments to the position he now vacates to assume his new duties.

♦

Isaac F. Kinnard, manager of engineering of the Meter and Instrument department of General Electric Co., was awarded an honorary doctor of science degree during recent convocation ceremonies at Queen's University, Kingston, Ontario, Canada. He was honored for his professional engineering accomplishments and his efforts to develop the engineers and scientists of the future. Dr. Kinnard is the author of "Successful Product Design," which appeared in the June 1953 issue of MACHINE DESIGN.

♦

C. C. Hurlburt has joined the Welding Div. of P. R. Mallory & Co. Inc., Indianapolis. Mr. Hurlburt was recently chief standards engineer for the Piasecki Helicopter Corp. and before that did procedure and engineering standards work for Fairchild Engine & Airplane Corp. and for the Curtiss-Wright Corp.

♦

Alten Foundry & Machine Works Inc., Lancaster, O., has announced the promotion of **Fred R. Bayne** to the position of chief engineer. Mr. Bayne came to Alten as assistant chief engineer from the Cincinnati Milling Machine Co. in 1951. Announcement was

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Want *real action* on your gear orders? You'll get it when you deal with Illinois Gear . . *action* that guarantees you highest quality and precision . . *action* that assures you on-time delivery . . *action* that offers you around-the-clock production for your gear emergency.

Illinois Gear, the World's largest, most modern, best equipped gear manufacturing plants, offers you the coordinated, synchronized action that makes a champion. It results from broad experience, second-to-none facilities and above all the will to produce!

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ILLINOIS GEARS Reg.U.S.Pat.Off.

Gears for Every Purpose ... one gear or 10,000 or more

ILLINOIS GEAR & MACHINE COMPANY

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also made of the appointment of **William A. Morris Jr.** as product development engineer. Mr. Morris has served as consulting engineer at Parkersburg Rig & Reel for the past ten years.

•

With duties that include overall responsibility for engineering and research, **Theodore W. Rundell** has been appointed vice president in charge of operations at Servel Inc., Evansville, Ind.

•

Victor Frank Tripoli, former design engineer with Gaynes Engineering Co., has joined the mechanism and dynamics department staff of Armour Research Foundation of Illinois Institute of Technology, Chicago. Mr. Tripoli will be a design engineer in the machine structure section of mechanism and dynamics.

•

Vice president in charge of engineering since 1951, **William H. Harris Jr.** was recently elected to the board of directors of Micromatic Hone Corp., Detroit.

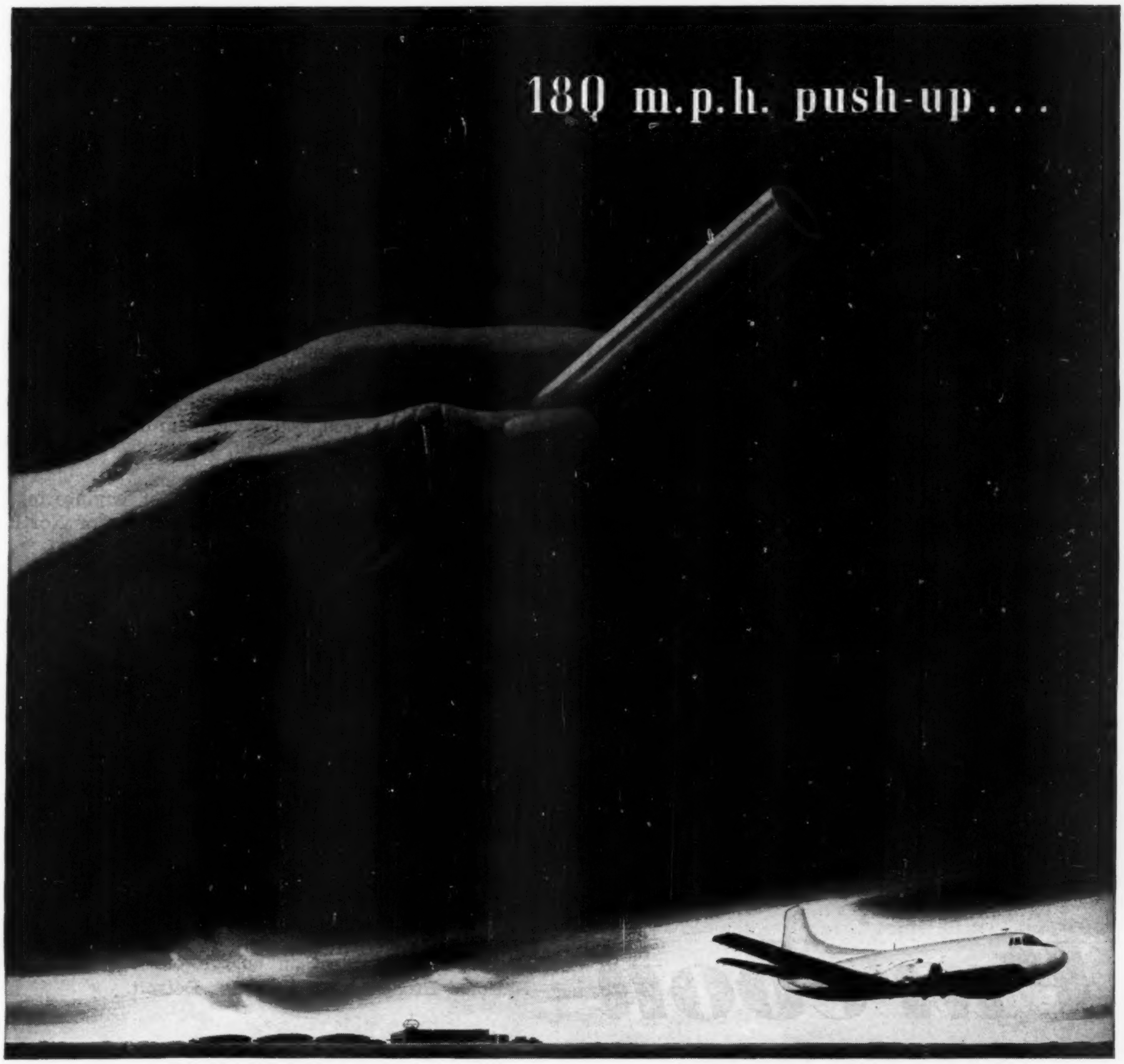
•

The Atomic Power Div. of Westinghouse Electric Corp., Pittsburgh, has announced the appointment of **William E. Shoupp** as assistant division manager in charge of development. Dr. Shoupp formerly served as director of development for the division.

•

E. W. Spannhake

E. W. Spannhake was recently named director of engineering and research of LeTourneau-Westinghouse Co., Peoria, Ill. He will be responsible for co-ordinating engineering functions involving product research, design and development. Mr. Spannhake, who was born in Germany, received his bachelor's and master's degrees in mechanical engineering at Massachusetts Institute of Technology. He engaged in private engineering consulting work both in the United States and abroad and was employed by American Bosch Co., American Locomotive Co. and the Lima-Hamilton Co. in various engineering and research capacities. For the past five years he has been associated with Barnes and Reinecke Inc. and served that firm as technical director of research and development activities. Mr.



Any time you fly in a Martin 4-0-4, Superior tubing is probably working for your safety.

Once you're airborne, watch the 840-pound main landing gear assemblies flip forward and up with amazing speed, even though they may be working against a 180 m.p.h. drag. Fast landing gear retraction gives you an extra margin of take-off safety because "clean" aircraft have better flight characteristics.

Chances are your Martin 4-0-4 contains many feet of Superior ½" stainless steel tubing. In the hydraulic system, this tubing operates at a pressure of 3000 p.s.i., normally. On Martin's torture racks, it has demonstrated remarkable endurance under violent pressure surges at 1000 cycles *per second*, even around minimum bends.

Performance like this may well have a bearing on your production problems as well as on your personal safety. Superior's long experience in fine tubing, backed by highly-developed production equipment and extensive research and testing facilities assures you of top-quality small tubing for doing tough jobs well. Outline your own production problems in a letter to us, right now—we'll send you complete information and the appropriate Data Memo by return mail. Address: Superior Tube Company, 2010 Germantown Ave., Norristown, Pennsylvania.

Round and Shaped Tubing available in Carbon, Alloy, and Stainless Steels, Nickel Alloys and Beryllium Copper.

Superior

THE BIG NAME IN SMALL TUBING

West Coast: Pacific Tube Company, 5710 Smithway St., Los Angeles 22, Calif. UNderhill 0-1331

All analyses .010" to ⅝" O.D.
Certain analyses (.035" Max. wall) up to 1⅜" O.D.

Spannhake is a member of the American Society of Mechanical Engineers and the Society of Automotive Engineers.

♦

Airway Products Inc., Pontiac, Mich., has appointed **Ernst F. Klessig** as director of engineering. Mr. Klessig was formerly executive design engineer at Vickers Inc.

♦

Robert G. Gustavson has joined the engineering department of Pastushin Aviation Corp., Los Angeles, as projects engineer in charge of jettisonable fuel tanks, external stores, and mechanism design.

♦

Raymond H. Rice, vice president of engineering, has been elected to the board of directors of North American Aviation Inc., Los Angeles.

♦

Wesley E. Bartholomew, W. K. Donaldson and **Saul M. Ferman** recently joined the technical staff of Hughes Research and Development Laboratories, Culver City, Calif.

♦

Perry Arant, who has headed product engineering activities at Clayton Mfg. Co., El Monte, Calif., for the past ten years, has been appointed vice president in charge of engineering. Mr. Arent joined the company's engineering staff in 1935.

♦

Formerly executive engineer, **Ben. F. Bregi** has been elected vice president in charge of engineering of National Broach & Machine Co., Detroit.

♦

Included among recent appointments made by Carboloy Department of General Electric Co., Detroit, is that of **David J. Jay** as an engineer in the product and process development section. Mr. Jay was previously associated with the aircraft nuclear propulsion project of the company's Aircraft Gas Turbine Div.

G. D. Barcus and **C. H. Good** recently joined the design and application engineering section of the Carboloy department. Mr. Barcus was formerly associated with the company's engineering services division in Schenectady, N. Y., and Mr. Good recently completed three months in the department's training program.

♦

Walter B. Claus, who has served as chief mechanical engineer of the Consolidated Engineering Corp., Pasadena, Calif., since 1949, has been named director of manufacturing of the firm's newly reorganized Transducer Div. He will also act as assistant head of the division. Succeeding Mr. Claus as chief mechanical engineer is **Gerald S. Perkins,** who has been assistant chief mechanical engineer since January.

THE ENGINEER'S Library

Recent Books

Refractory Hard Metals. *By Dr. Paul Schwarzkopf and Dr. Richard Kieffer; 459 pages, 6 by 9 inches, clothbound; published by the MacMillan Co., New York; available from* MACHINE DESIGN; *$10.00 postpaid.*

Devoted primarily to metallurgical considerations, this book deals with the preparation and properties of refractory and hard carbides, nitrides, borides, and silicides of transition metals. It also includes chapters embracing the application of these substances as constituents of high-temperature materials.

•

Timing Engineering. *By Myrten G. Saake, application engineer; 6 by 9 inches, 255 pages, clothbound; published by the Ribble Engineering Co., Jersey City 2, N. J., available from* MACHINE DESIGN; *$5.00 postpaid.*

Basic design, construction and application of timers in relation to automatic control of machinery and production processes make up the contents of this volume. Timers are covered in their simplest to the most complex forms, and many of their applications are liberally illustrated with circuit diagrams and photographs. Several techniques are shown for combining timers with counters to achieve automatic control.

•

Chemical Process Machinery. *By E. Raymond Riegel, Professor emeritus of Chemistry, University of Buffalo; 6 by 9 inches, 743 pages, clothbound; published by Reinhold Publishing Corp., New York; available from* MACHINE DESIGN; *$12.50 postpaid.*

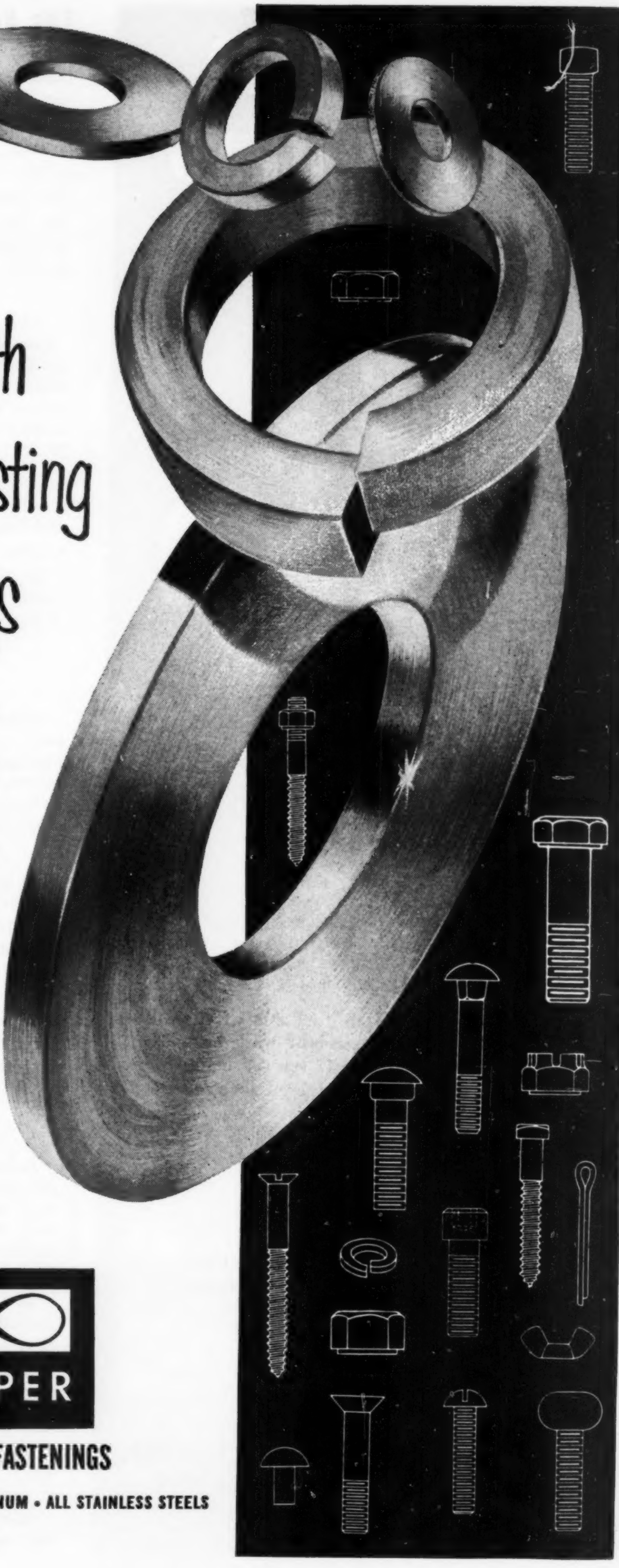
This enlarged version of Riehel's *Chemical Machinery* brings the material of the previous volume up-to-date with latest developments of equipment employed by chemical and process industries. There is new material in this second edition on crushers, magnetic separators, centrifuges, turbodriers, and other process equipment.

•

Pumps. *By Frank A. Kristal, mechanical engineer, and F. A. Annett, contributing editor, "Power"; 391 pages, 6 by 9 inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, *$6.50 postpaid.*

A comprehensive discussion of different types of

end corrosion troubles with HARPER everlasting fastenings



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pumps available, their selection, installation, operating characteristics, and maintenance is given in this second edition. Basic pumps covered include reciprocating; volute and diffuser centrifugal; turbine; propeller; rotary; deep-well; boiler-feed; sewage, sludge, and sanitary; variable displacement and proportioning types. Sixteen tables of data are included in the appendix to aid in the selection and application of pumps.

♦

Cambridge Elementary Statistical Tables. *By D. V. Lindley and J. C. P. Miller, Cambridge University; 36 pages, 8½ by 11 inches, paperbound; published by Cambridge University Press; Available from Cambridge University Press, American Branch, 32 E. 57th St., New York 22, N. Y., $1.00 per copy.*

Common statistical functions, tests of significance, and functions of integers make up the contents of this booklet. Tables include normal, t, X^2, and F distributions. For the latter three, percentage points are provided for significance levels of 5, 1, and 0.1 per cent in both one-sided and two-sided tests.

♦

Elements of Heat Treatment. *By George M. Enos, late professor of metallurgical engineering, and William E. Fontaine, associate professor of mechanical engineering, Purdue University; 294 pages, 6 by 9 inches, clothbound; published by John Wiley and Sons Inc., New York; available from* MACHINE DESIGN, *$5.00 postpaid.*

Basic principles governing the application of heat to alter the mechanical properties of metals are well illustrated and described in this book. Following sections on mechanical properties and working of metal there is a discussion of the major heat treatment processes—annealing, hardening, tempering, and normalizing. Casting, forging, rolling, welding, machining and other processes related directly or indirectly to heat treatment are briefly covered.

♦

Industrial Specifications. *By E. N. MacNiece, director of quality control, Johnson & Johnson Co., 172 pages, 6 by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN; *$4.50 postpaid.*

Methods of planning, writing, and issuing specifications for the various phases of an industrial operation are given in this book. The materials include a discussion of variability and tolerances, standards and standardization, and procedures and requirements for raw material, process, and product specifications.

♦

Metallurgical Dictionary. *By J. G. Henderson, consulting engineer, and J. M. Bates, metallurgical engineer;*

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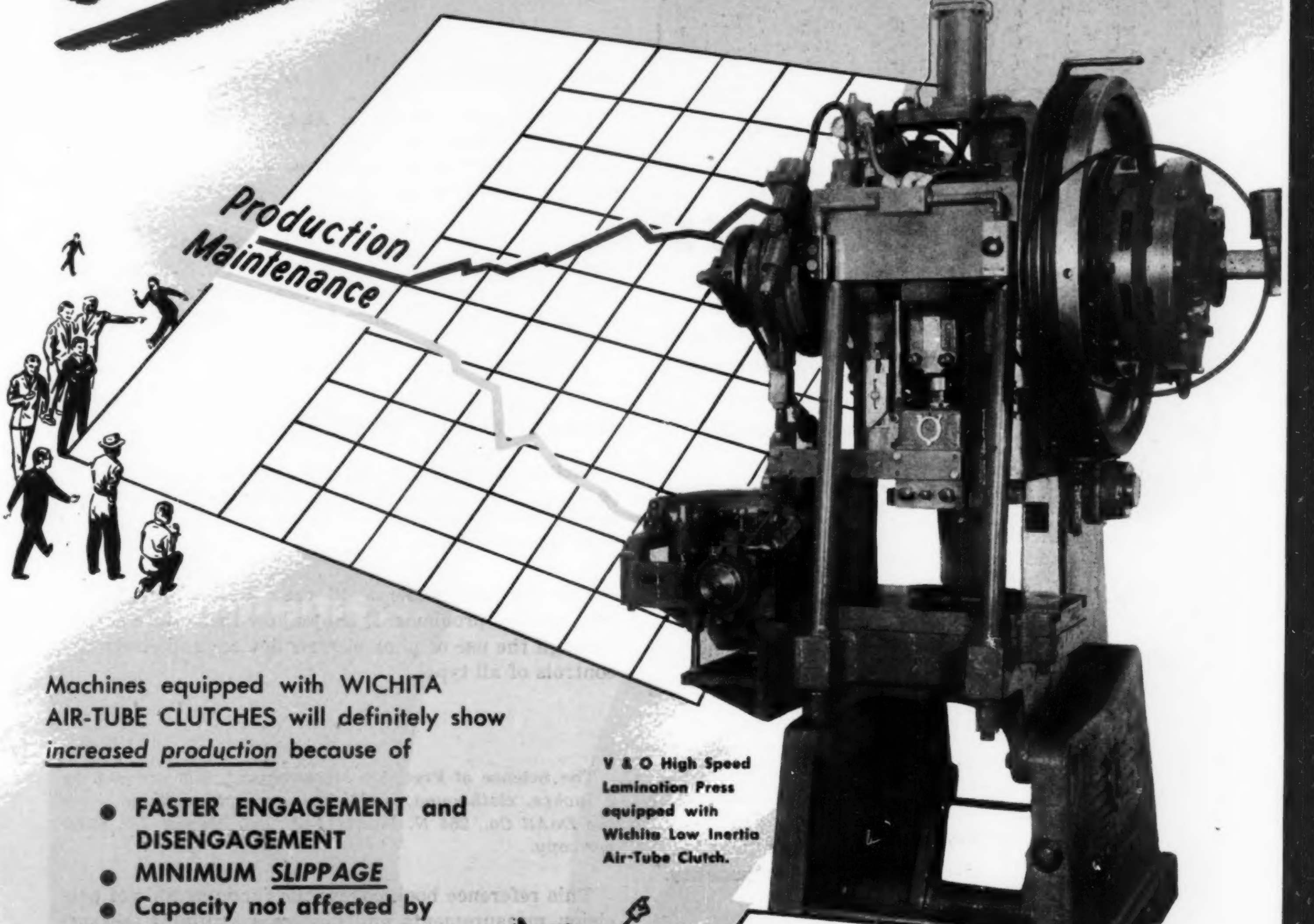

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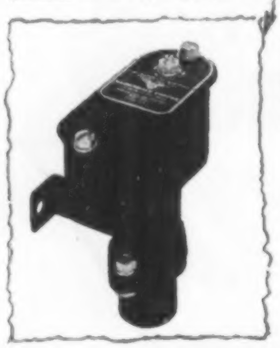


The Engineer's Library

408 pages, 6 by 9 inches, clothbound; published by Reinhold Publishing Corp., New York; available from MACHINE DESIGN, *$8.50 postpaid.*

Written primarily for men working in the metallurgical and associated fields, this reference volume contains over 5000 definitions and descriptions covering most of the essential terms in both production and physical metallurgy. Related terms are cross referenced to direct the user to preferred terms.

New Standards

Letter Symbols for Radio. *ASA Y10.9-1953; 10 pages, 8½ by 11 inches, paperbound; available from American Society of Mechanical Engineers, 29 W. 39th St., New York 18, N. Y., $1.00 per copy.*

General principles of letter symbol standardization for radio introduce this standard. Six pages of the material are devoted to two alphabetical lists—one in order of quantity and the other in order of symbol.

Manufacturers' Publications

Cutting Production Costs with Electronic Controls. *76 pages, 5¼ by 7½ inches, paperbound; available from Photoswitch Inc., 77 Broadway, Cambridge 42, Mass., on company letterhead request.*

This booklet presents 46 case histories of industrial control problems. It shows how these were solved through the use of photoelectric devices and electronic controls of all types.

•

The Science of Precision Measurement. *258 pages, 6 by 9 inches, clothbound; published by and available from the DoAll Co., 254 N. Laurel Ave., Des Plaines, Ill., $3.50 per copy.*

This reference book covers the fundamentals of precision measurements and discusses available measuring devices. Clearances and tolerances, gage blocks, angle measurement, electrical and mechanical comparators, and thread and gear measurement are among the subjects included.

Government Publications

NACA Technical Series. *Each publication is 8 by 10½ inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics, 1924 F St. N. W., Washington 25, D. C.*

The following Technical Notes are available:

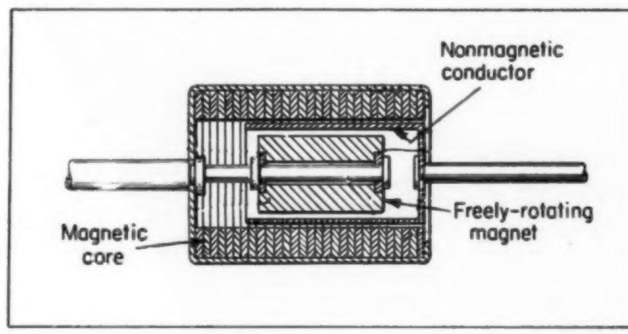
3027. **Influence of Rotor-Engine Torsional Oscillation on Control of Gas-Turbine Engine Geared to Helicopter Rotor**—40 pages.

3042. **High-Frequency Pressure Indicators for Aerodynamic Problems**—52 pages.

NOTEWORTHY

Patents

RESTRICTED TORQUE TRANSMISSION for small mechanisms, regardless of direction of rotation or speed, is achieved magnetically by a coupling detailed in patent 2,636,139. Designed by James L. Winget and assigned to Farrand Optical Co., the device may be employed as a rotational damper and transmits a torque proportional to and controlled by the difference in velocities between the driving and driven members. Operation of the unit is based on the interaction of magnetic and electrical forces produced by an assembly consisting of a cylindrical nonmagnetic conductor sleeve connected to the driving shaft and mounted to fit between a freely rotating inner



magnet and an outer magnetic core fastened to the driven shaft. Rotation of the conductor sleeve generates eddy currents, imposing a torque on the magnet proportional to the difference in velocities which is transmitted to the outer core by magnetic attraction. Hysteresis forces limit the power transmitted, permitting the magnet to rotate relative to the core when a certain torque value is exceeded. Maximum torque as well as minimum velocity difference are a function of the size, shape, materials, and spatial relationships of the three members. Two modifications, one for an axial, rather than radial, arrangement of rotating members and the other for a linear motion device, are also shown.

COMBINATION PULLEY-CLUTCH DESIGN has automatic wear takeup which simplifies adjustment and maintenance of a unit described in patent 2,639,-015. Assigned to Twin Disc Clutch Co., the design consists of a free-running sheave and a friction clutch mounted to the driven shaft as a single unit. Clutching action is provided by balls mounted in tapered pockets, which are actuated by a movable collar to drive a keyed friction disc into engagement with one side of the sheave. In operation, the clutch surfaces are spring-loaded during engagement, automatically

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Like the gentleman-on-a-picnic, above, you're scheduled for some pretty fast action yourself. Because, for all business, the day's coming when materials become more plentiful, as restrictions disappear. That's also the day when orders will be harder to find, with competition as tough as any charging bull could possibly be. ● Give yourself a competitive edge with stainless steel. Nothing else is as hard, strong and lastingly attractive—as impervious to corrosion, heat and wear. Used in equipment, Allegheny Metal can save on your costs . . . used in a product, it can add vital sales qualities. Now's the time to plan with it and for it, and all our facilities are at your command. *Allegheny Ludlum Steel Corporation, Oliver Building, Pittsburgh 22, Pa.*

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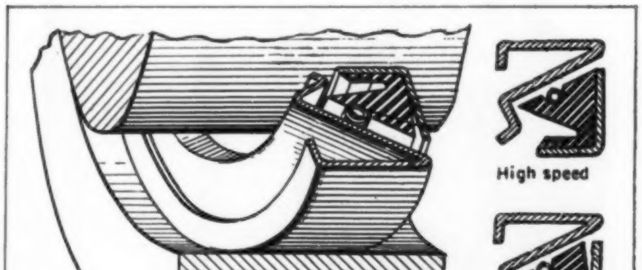
compensating for wear and providing a uniform friction force. Smooth release and pickup of load are obtained by the design which also locks automatically in the engaged or disengaged position. A modification employing rollers instead of balls is also shown. Inventor Lloyd J. Wolf is the assignor of the patent.

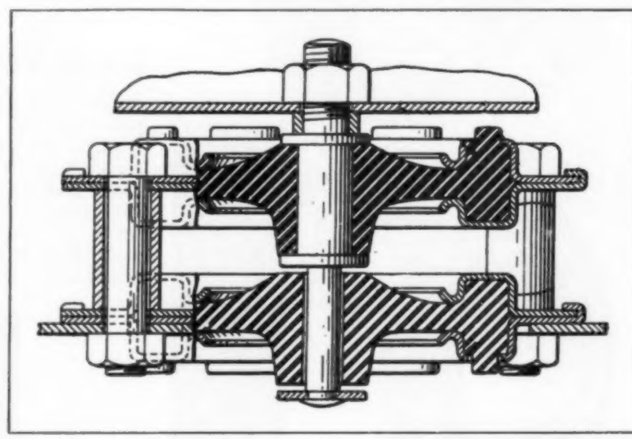
FRICTIONLESS SEALING at high speeds is obtained with a novel shaft seal which operates effectively at low speeds or at a standstill. Described in patent 2,637,574, the seal has been designed for use with a shaft and housing and consists of two separate press-fitted units; a Z-shaped metal ring mounted in the housing and a metal-reinforced rubber sealing element, encircled by a tension spring, on the shaft. At low shaft speeds or at a standstill, sealing action is provided by positive engagement of the sealing element with the metal ring, and is maintained by the tension spring. Above a certain critical speed, however, centrifugal force overcomes the spring force, disengaging the mating surfaces and introducing a frictionless slinger sealing action. Pockets in the sealing element act as fluid slingers to create a turbulent condition and block flow past the seal. Installation is facilitated by the two-piece design which keeps frictional contact to a minimum and reduces wear and temperature build up. Inventor Karl L. Diehl has assigned the patent to Victor Mfg. & Gasket Co.



CUSHIONED LINEAR MOTION for positioning or control is obtained by means of a new telescoping tube actuator which affords increased rigidity. Detailed in patent 2,639,625, the actuator designed by Howard M. Geyer and Donald K. Ferris employs a ball-bearing screw and nut drive, driven by a reversible electric motor, to extend and retract a tubular member mounted to the nut and sealed at one end by a check valve. Power connections are made to the free end of the tube by means of a clevis or mounting block. Sealing and rigidity of the unit are provided by a stationary tubular chamber, enclosing the screw

drive, in which the actuator tube slides. In operation, entrapped air is utilized to cushion and retard the motion of the actuator tube at its travel limits. Excessive pressure buildup on the retracting stroke is relieved through the check valve. Assignee of the patent is General Motors Corp.

LIMITED VIBRATION AMPLITUDE at or near resonant frequency minimizes transmitted energy in the antivibration mount covered in patent 2,639,115. Designed to provide a soft support for instruments and similar fragile equipment, the mount assigned to General Tire and Rubber Co. employs two doughnut-shaped resilient rubber elements, complementing each other in action, to isolate both high and low-frequency vibrations. Under normal conditions and low amplitudes, damping action is provided by one of the ele-



ments operating independently. As resonant conditions are approached, however, the resulting increase in amplitude actuates the second resilient element, changing the natural frequency of the mount. This action increases the effective stiffness to vibration and prevents further amplification while at the same time acting to reduce the existing amplitude. A modified version of the device is also shown in the patent which has been assigned by inventor Robert Iredell, Jr.

AUTOMATIC PRESSURE RELIEF to reduce pumping loads in hydraulic circuits employing accumulators is provided by an unloading valve assigned to Bendix Aviation Corp. Designed by Walter D. Teague Jr. and detailed in patent 2,639,722, the valve is used between the pump and accumulator and can be preset to unload the pump when a prescribed pressure is built up. Under normal operating conditions, fluid flows through the valve from the pump to the accumulator. When the pressure build-up reaches a specified value, a small pilot piston actuates a plunger to divert flow through a drain port, relieving the pressure load on the pump and reducing power require-

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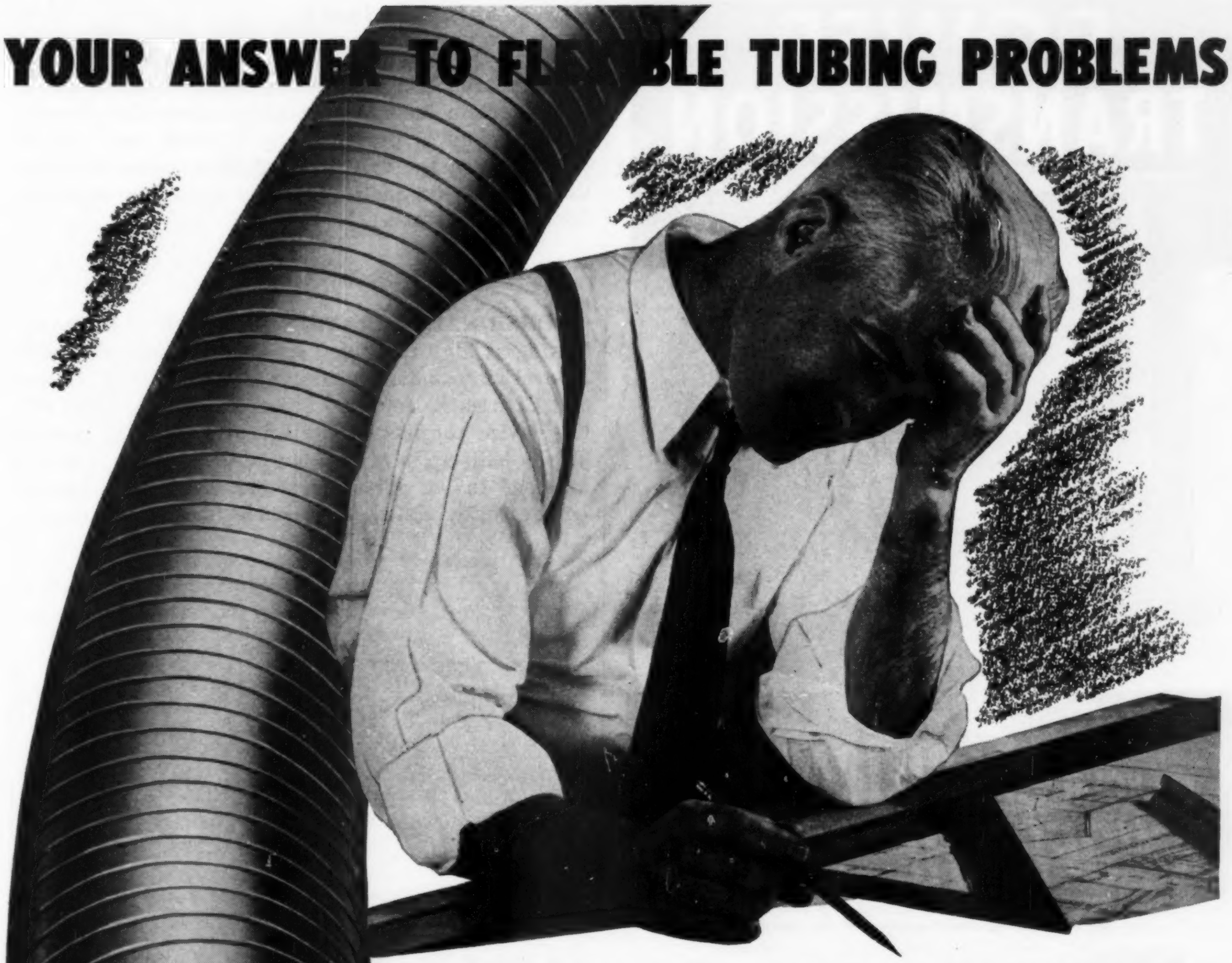

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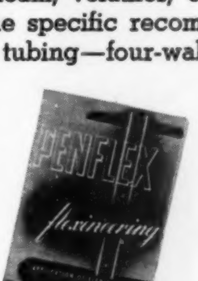

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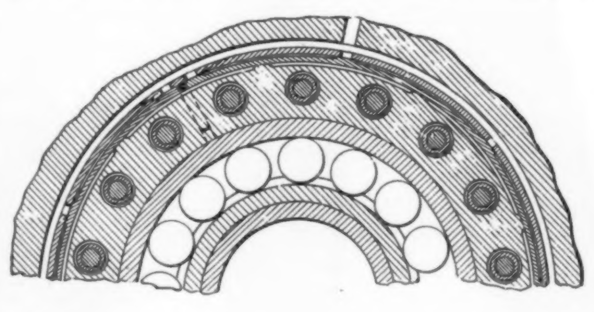
ments. Reduction of pressure in the accumulator below a certain limit automatically resets the plunger. Fluid backup is prevented by a check valve in the outlet side of the valve. In addition, valve construction assures adequate operating forces to overcome frictional resistance.

TORSIONAL VIBRATION DAMPING of shafts operating at high rotational speeds is accomplished with a novel viscous damper designed by Martin M. Holben and Charles S. Jewett. Covered in patent 2,631,901, the device consists of a stationary resilient circular sleeve, built-up in layers of ribbed segments, which is attached to the outer race of an antifriction bearing on the shaft. Oil in the pockets formed between the



ribs acts as the damping medium. Lateral vibrations of the shaft deflect the sleeve members, forcing the oil along a tortuous path through the ribs to provide a damping action. Continual circulation of the oil to remove heat generated during operation is provided by an external pump. The patent has been assigned to Curtiss-Wright Corp.

SPEED-RESPONSIVE CONTROL of electrical switching is provided by a modified flyball governor mechanism designed by Harry E. Sloan. Assigned to Cushman Chuck Co., the device covered in patent 2,639,135, opens and closes a limit switch automatically in response to linear or rotational speed variations. Switch position is controlled by ball cams, mounted to roll back and forth in inclined radial tracks in a revolving rotor, which move a spring-loaded plate up and down to actuate a bellcrank switch arm. With the rotor stationary, the balls are pulled toward the center of rotation by gravity, raising the plate and opening the switch. With the rotor in motion, centrifugal force drives the balls outward, lowering the plate and closing the switch. Open and closed positions of the switch may be interchanged to meet design requirements. In addition, sensitivity of response is adjustable through a stud on the top of the plate. A modification providing a reversed plate action is also shown.

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Cutaway shows compact interior design and construction of a typical Cleveland worm gear speed reducer, used in thousands of plants. Clevelands save floor space—are simple to install and maintain. A minimum number of moving parts practically eliminates trouble.

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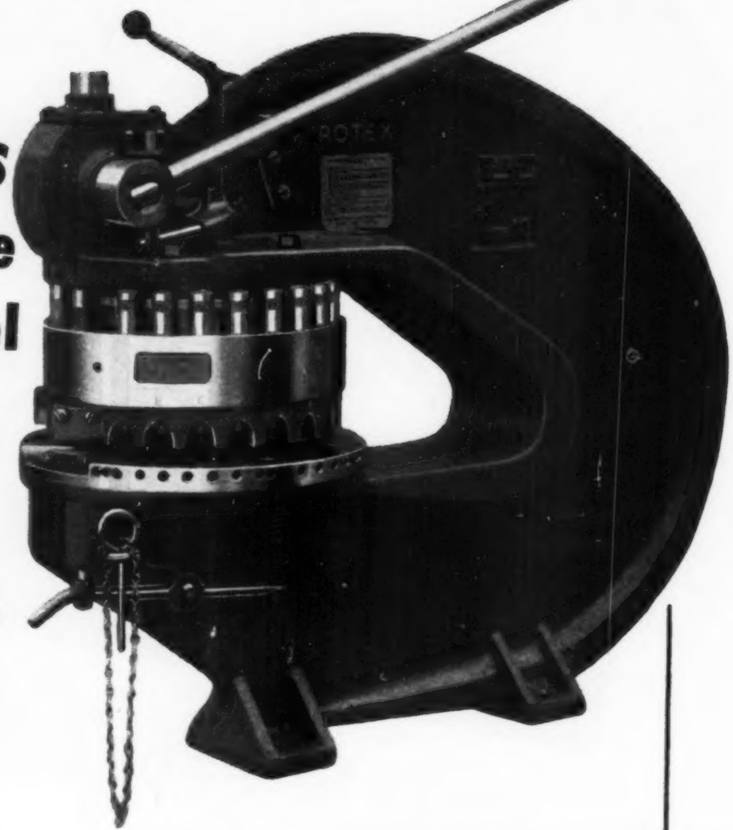
Die Turret Plate



Speed Case (X1515) low carbon, open hearth, steel plate has won the favor of numerous manufacturers of machine tools, because of its machinability, its ease of finishing and response to the various case hardening methods.

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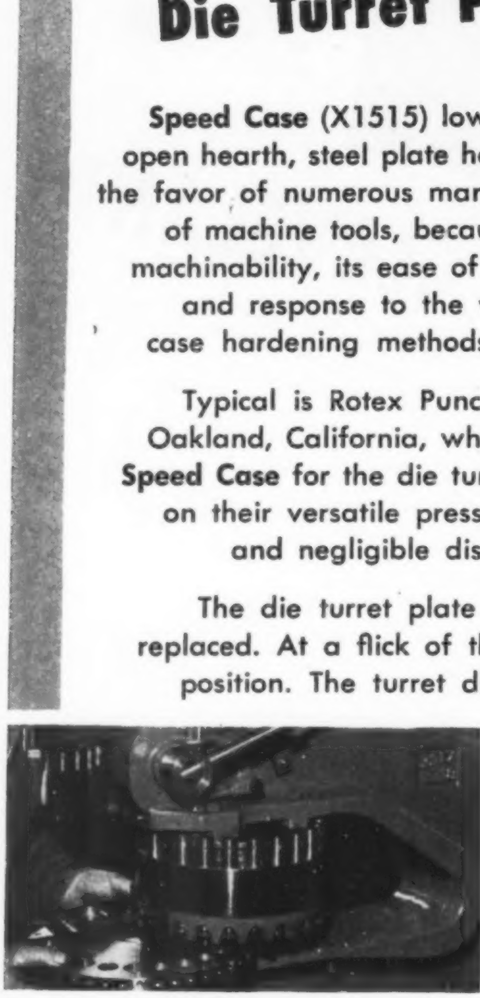
The die turret plate retains the dies which are readily removed and replaced. At a flick of the wrist a different size punch can be brought into position. The turret die plate is then rotated to desired position and locked in place. Rotex "Quick Change" Punch Presses are widely used in sheet metal, fiberboard, plastic and other fabricating plants. Punches and dies may be round, square or other shapes. **Speed Case** has ideal physical properties for this application. Perhaps it can save time and cut costs on your equipment.



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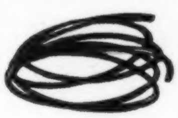

that answers a centuries'-old search. It's light in weight, won't rust or corrode, is smooth-walled for full flow, and easy to install.

BAKELITE *Polyethylene* started a revolution in packaging with the unbreakable squeeze bottle dispensing everything from cosmetics to insect sprays, shampoos to lubricants, car washes to medicines . . . in colorful, styled packaging with dramatic sales and utility appeal.

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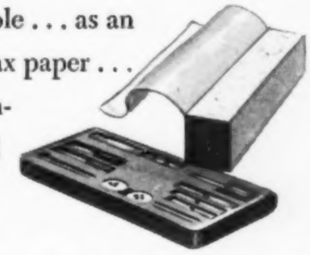

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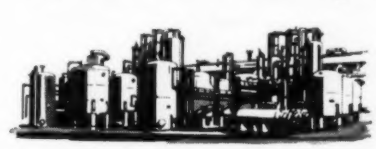

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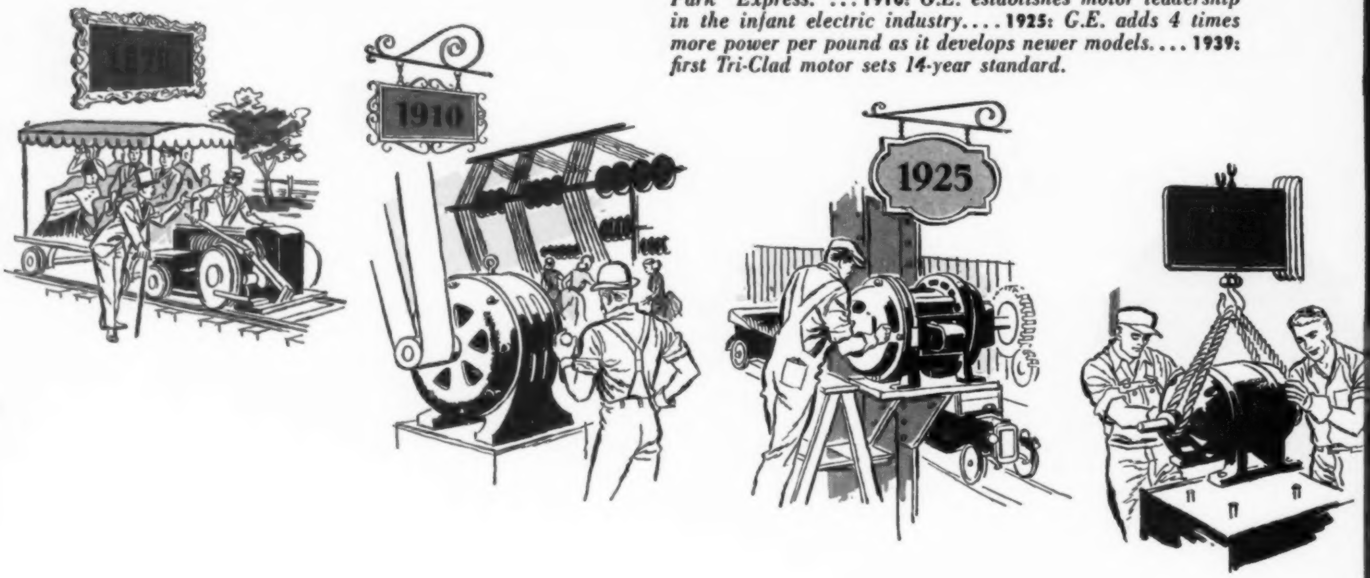

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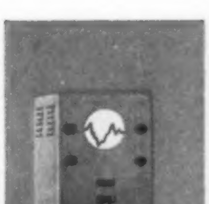
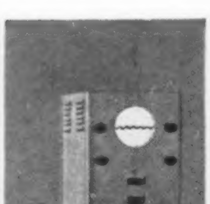
REG U S PAT OFF

The General Electric Tri/Clad '55' is an important milestone in motor history. For this is a *completely new motor*. Born out of thousands of engineering man-hours, the Tri/Clad '55' incorporates design improvements that go far beyond mere modifications. Many years of pure research, the discovery of new, better materials, the knowledge of how to make better use of present materials, and improved manufacturing processes all make the Tri/Clad '55' motor the new leader in the motor field.

You as a motor user should take the opportunity to see and test this motor for yourself. Contact your nearest G.E. Apparatus Sales Office or G.E. Motor Agent and tell him you want to see the new G-E Tri/Clad '55'. General Electric Company, Schenectady 5, N. Y.

648-4

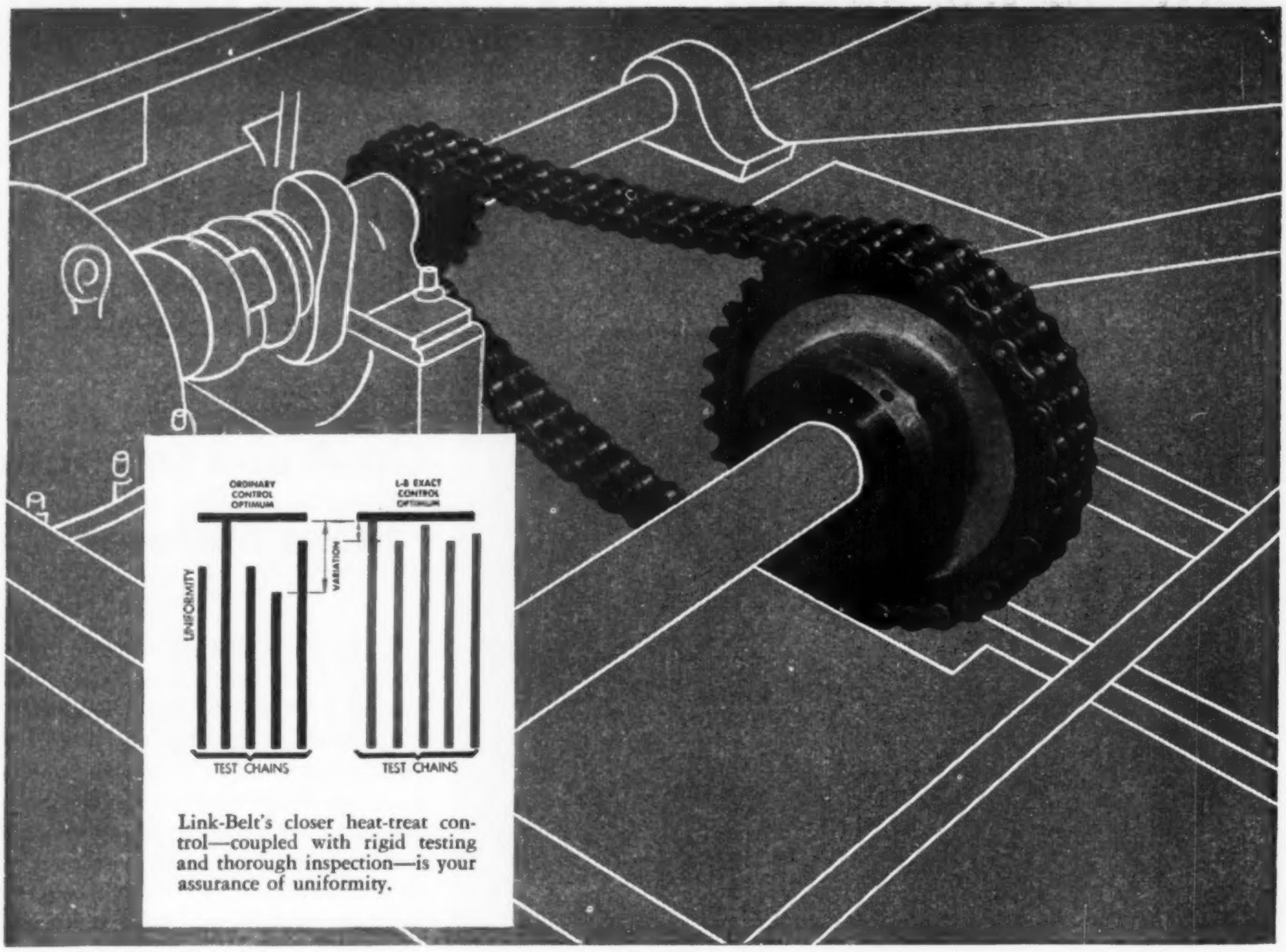




DESIGNED FOR BETTER PERFORMANCE: Tri/Clad '55' has a low noise level ... runs cool ... gives up to 53% increase in shaft output per pound...has higher full-load speeds.

ENGINEERED FOR LESS MAINTENANCE: 10%-life tests (such as the one shown) prove the Tri/Clad '55' bearings last longer without regreasing than any other bearing system.

BUILT TO LAST LONGER: cast-iron strength . . . more fully enclosed dripproof construction . . . water-shedding stator windings ... polyester film insulation 8 times stronger.



Link-Belt's closer heat-treat control—coupled with rigid testing and thorough inspection—is your assurance of uniformity.

Closer heat treat control means longer roller chain life!

But it's only one of many engineering extras you get from LINK-BELT

THE long-life record of Link-Belt Precision Steel Roller Chain is based on more than closer heat treat control. Consider such extra-wear features as shot-peened rollers and lock-type bushings. They're further assurance of Link-Belt's *built-in* extra life.

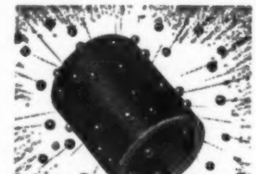
Link-Belt Roller Chain is available in single or multiple widths, 3/8" through 3" pitch. Double pitch, 1" through 3". For all the facts, call the Link-Belt office near you today, or write for Data Book No. 2457.

LINK-BELT

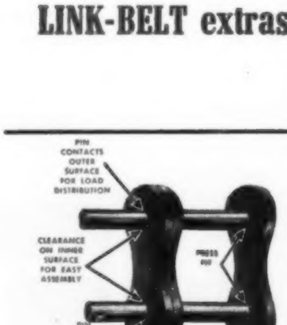
ROLLER CHAIN & SPROCKETS

LINK-BELT COMPANY: Plants: Chicago, Indianapolis, Philadelphia, Colmar, Pa., Atlanta, Houston, Minneapolis, San Francisco, Los Angeles, Seattle, Toronto, Springs (South Africa), Sydney (Australia). Sales Offices, Factory Branch Stores and Distributors in Principal Cities. 13,120

Don't overlook these other LINK-BELT extras



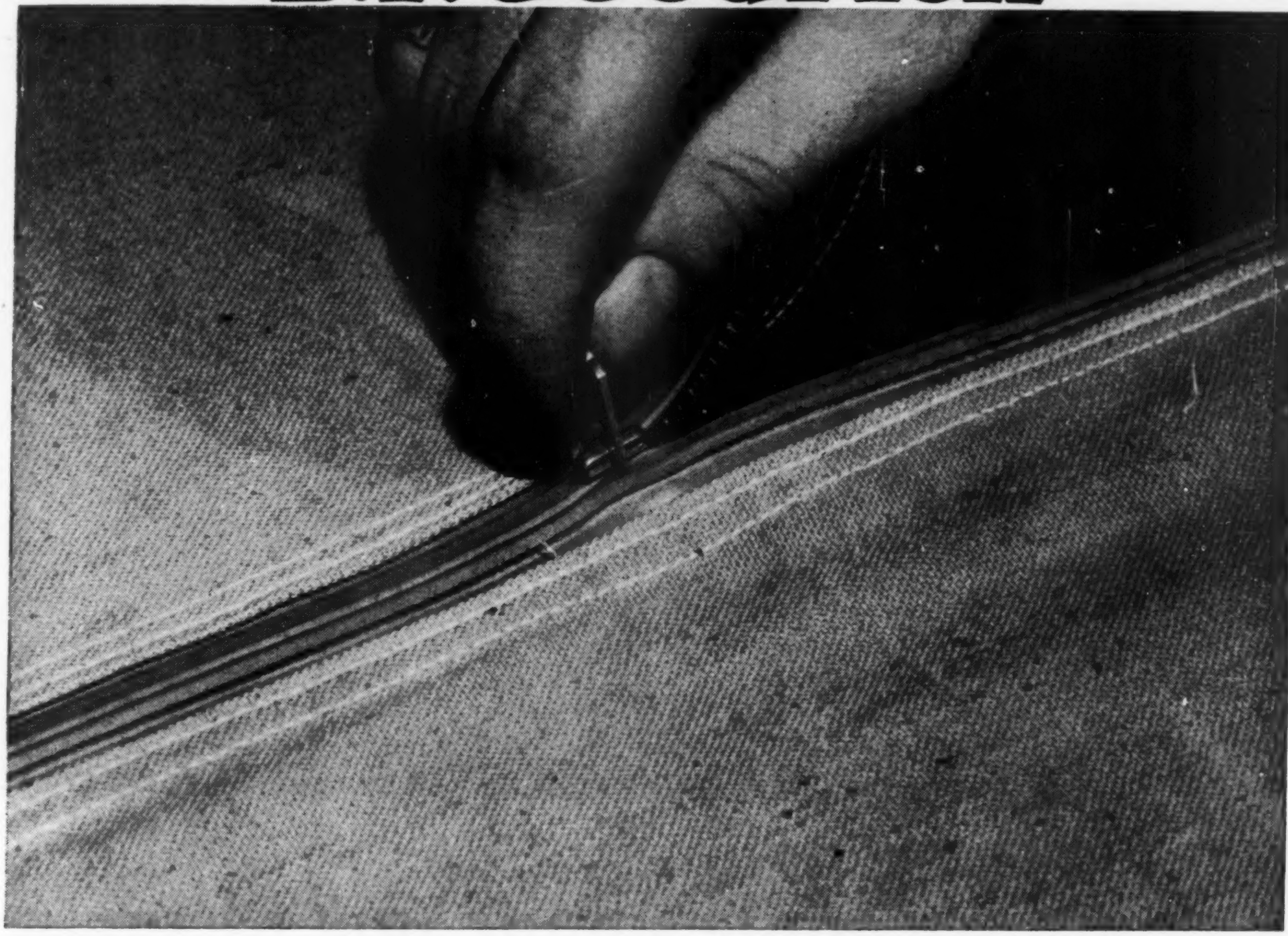
Shot-peened rollers have extra fatigue life for higher speeds, heavier loads.



Couple, uncouple multiple-width chains easily.

Lock-type bushings assure free articulation.

RESEARCH KEEPS **B.F.Goodrich** FIRST IN RUBBER



New kind of seal is watertight, airtight...zips open fast

A NEW KIND OF SEAL keeps gases, liquids, dirt *in* or *out*. Here's how it works:

Rubber lips are molded with hairline precision so that they fit together tightly. They press together even without pressure to seal effectively. And with pressure against the lips, the seal actually tightens. The rubber seal is applied to an ordinary zipper which zips open or shut in a hurry.

The B. F. Goodrich Sealing Zipper is so flexible it goes around curves and odd shapes where clamps won't work.

Can be made of compounds to resist weather, oils, gasoline, chemicals. Can be sewn or cemented to fabric, metal, wood, glass. The rubber can be made to match any color. Production costs are often reduced because the zipper is so easy to install.

It has been used in life-saving suits, to seal delicate instruments against moisture, to seal lifeboats against sea water, to seal auto convertible tops against rain. It is ideal for any use that combines a need for an airtight, watertight seal with a need for quick opening.

Write for complete details, sending blueprints or specifications of your products to Dept. A-88 or send coupon below. The B. F. Goodrich Company, Zipper Division, Akron, Ohio.

Send Now for Free Folder

Gives case histories, design advantages, suggested uses.

The B. F. Goodrich Company

Dept. A-88, Akron 18, Ohio

Name ....

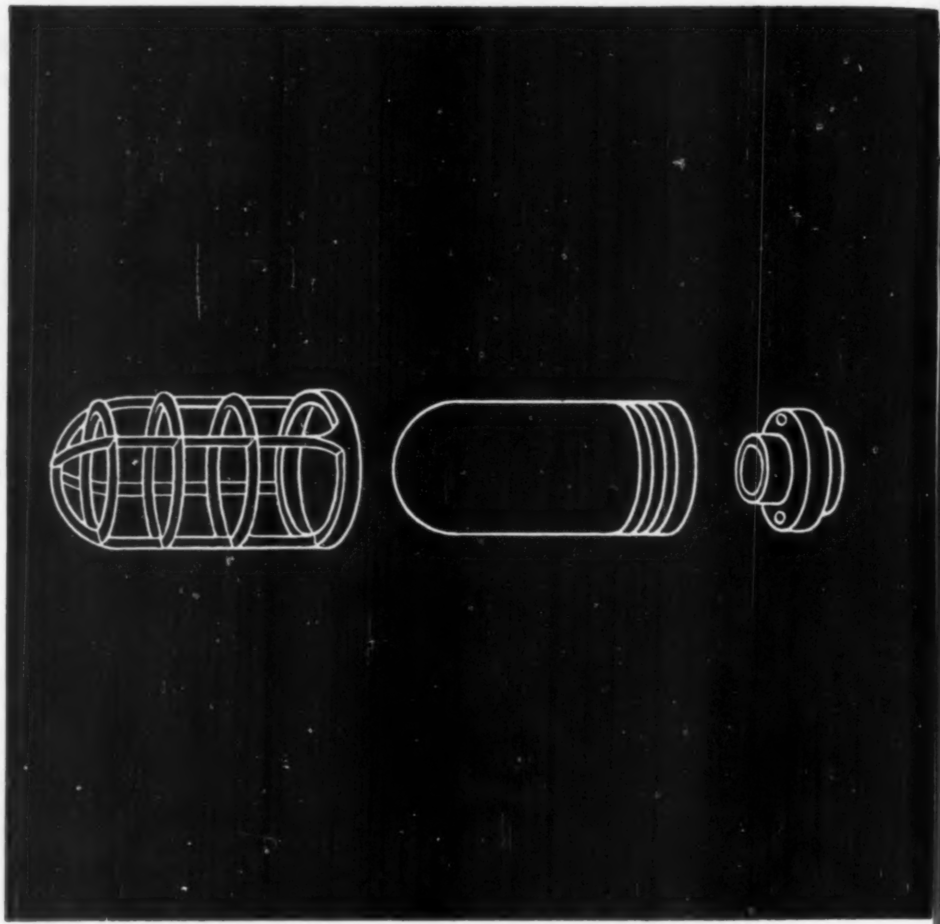
Firm ....

Title ....

Address ....

City .... *Zone* .... *State* ....

B.F.Goodrich **Airtight, Watertight Sealing Zipper**



NEW FIBER GASKET

...seals vapor out of lamp,

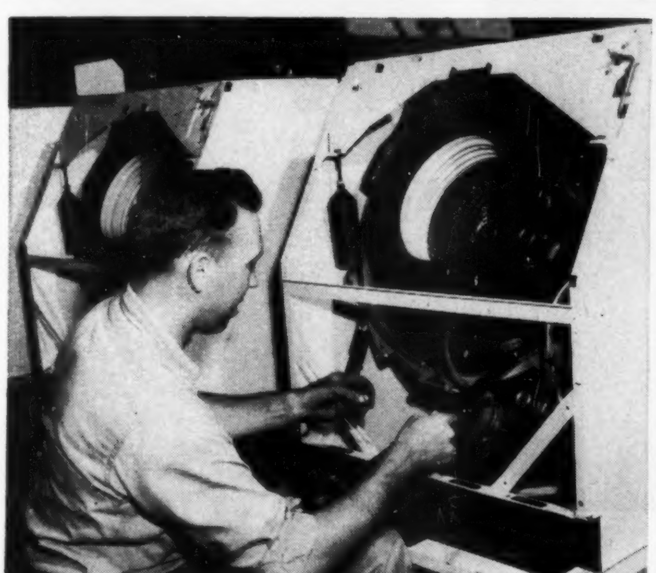
NEVER SHRINKS OR DRIES OUT. Both these gaskets were used for six months in a gear pump circulating hot hydraulic oil. The plant fiber gasket (left) lost its glycerine plasticizer, dried out, shrank, became hard, brittle. The Accopac gasket (right) remained virtually unaffected, as the photograph clearly shows.

Investigation of field complaints showed a lamp manufacturer he had an unusual gasket problem. His gaskets, which were doing a good job of keeping water vapor out, were nonetheless causing considerable dimming. The 350° F. heat of the lamp vaporized the binder in the gaskets. The extracted binder formed a dark deposit at the base of the lamp cover.

To correct this, he tried an entirely new kind of fiber gasket material—Armstrong's Accopac®. Severe testing proved that Accopac filled both his basic requirements—it made a vapor-tight joint under the low pressure of hand-tightened nuts, and its binder stayed in the gasket.

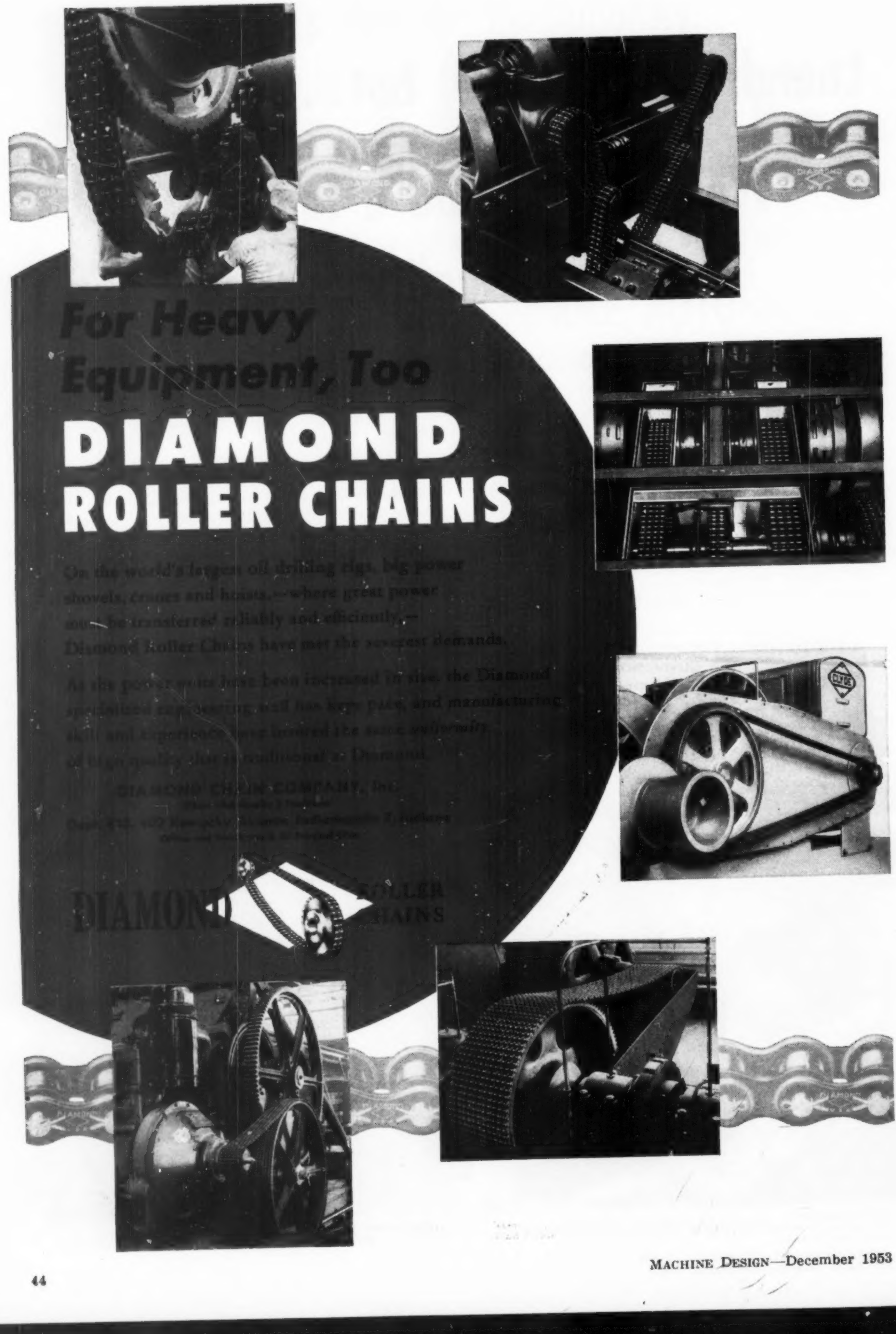
Won't shrink or dry out. Accopac contains only one binder—non-extractable latex. Unlike most fiber materials, Accopac is not made by dip-saturation, but by a patented variation of the beater saturation process. For this reason, Accopac gaskets won't lose their

ARMSTRONG'S



BOTTOM DAMPER springs are installed. Because of the entire engineered suspension, Laundromat was one of the first washers that didn't have to be bolted to floor.

WESTINGHOUSE LAYS DOWN rigid specifications for their American Quality Springs, proving their motto, "Quality must be *built* into a product."


For Heavy Equipment, Too
DIAMOND ROLLER CHAINS
On the world's largest oil drilling rigs, big power shovels, cranes and hoists,—where great power must be transferred reliably and efficiently,—Diamond Roller Chains have met the severest demands.
As the power units have been increased in size, the Diamond specialized engineering skill has kept pace, and manufacturing skill and experience have insured the same uniformity of high quality that is traditional at Diamond.
DIAMOND CHAIN COMPANY, Inc.
DIAMOND ROLLER CHAINS

PARKER-KALON®

originator and world's largest producer of hardened Self-tapping Screws, and a leading manufacturer of Socket Screws and other fasteners, is now a Division of General American Transportation Corporation

GENERAL AMERICAN, with principal offices in Chicago, owns and operates the nation's largest private fleet of freight cars which are leased to many of the foremost industrial companies, and to railroads. In addition, it is a major builder of all types of freight cars, owns and operates tank storage facilities, and is engaged in the manufacture of process equipment and molded plastics.

In pursuance of a program of diversification, General American has acquired the business and assets of the Parker-Kalon Corporation through the purchase of capital stock. The business will henceforth be operated as the Parker-Kalon Division of General American Transportation Corporation.

The present management of Parker-Kalon will continue in charge. The sales and service policies which have won the approval of the many thousands of P-K customers will remain in effect. As always, P-K Fasteners will be sold through P-K Distributors in every industrial area in the United States, Canada and in many foreign countries.

With the far greater resources available through its new ownership, Parker-Kalon will accelerate a program of improvement and expansion. New production and warehouse facilities will be built to meet the constantly increasing demands for P-K fasteners. Engineering and research will be augmented to speed the perfection of new P-K products now in development.

Now, even more than before, you can rely on Parker-Kalon for leadership in fastener design, for consistently high fastener quality, and service in step with your needs.

P-K TRADE MARK REG. U.S. PAT. OFF.

PARKER-KALON®

Division

General American Transportation Corporation

200 Varick Street, New York 14, N. Y.

GENERAL AMERICAN

Four different types of synchronous motor driven INTERVAL TIMERS

the answer to most interval timing-control problems

MANUAL SET TIMERS • Series RS

Compact, rugged construction handles high load without auxiliary relay. Heavy-duty, snap-action contacts maintain continuous pressure and wiping action. 18" color-coded leads facilitate wire connections. 20-amp. rating—completely enclosed—2½" dia., 3½" length — easy-setting knob — elapsed-time indication. Time-cycle ranges, 15 min. to 12 hrs., dial calibration, 15 sec. to 15 min. 115-volt AC current, 60, 50, or 25 cycles. *Complete information in Bulletin #59*

INSTANTANEOUS RESET TIMERS • Series PAB

For process control in industry. Have automatic reset, time-setting adjustment, large numerals. Extremely accurate. Built to stand hard usage. Timing range 1/10 sec. to 5 sec. in 1/10-sec. steps (PAB-3S)—to 2 min. to 3 hrs. in 2-min. steps (PAB-3H). *Complete information in Bulletin #58*

AUTOMATIC RESET TIMERS • Series P and M

Series P have AC input line cord, built-in actuating start button, receptacles for plug-in remote-control and load circuits. Time cycle, 15-sec. to 5-min., dial calibration, ¼ sec. to 5 sec. 115 or 220 volt AC, 50 or 60 cycles.

Series M is similar to Series P, but start button cannot operate from remote position. Receptacle for plug-in load circuit. 60-sec. time cycle, calibrated in seconds. *Complete information in Bulletin #68*

SIGNALING TIMERS • Series S

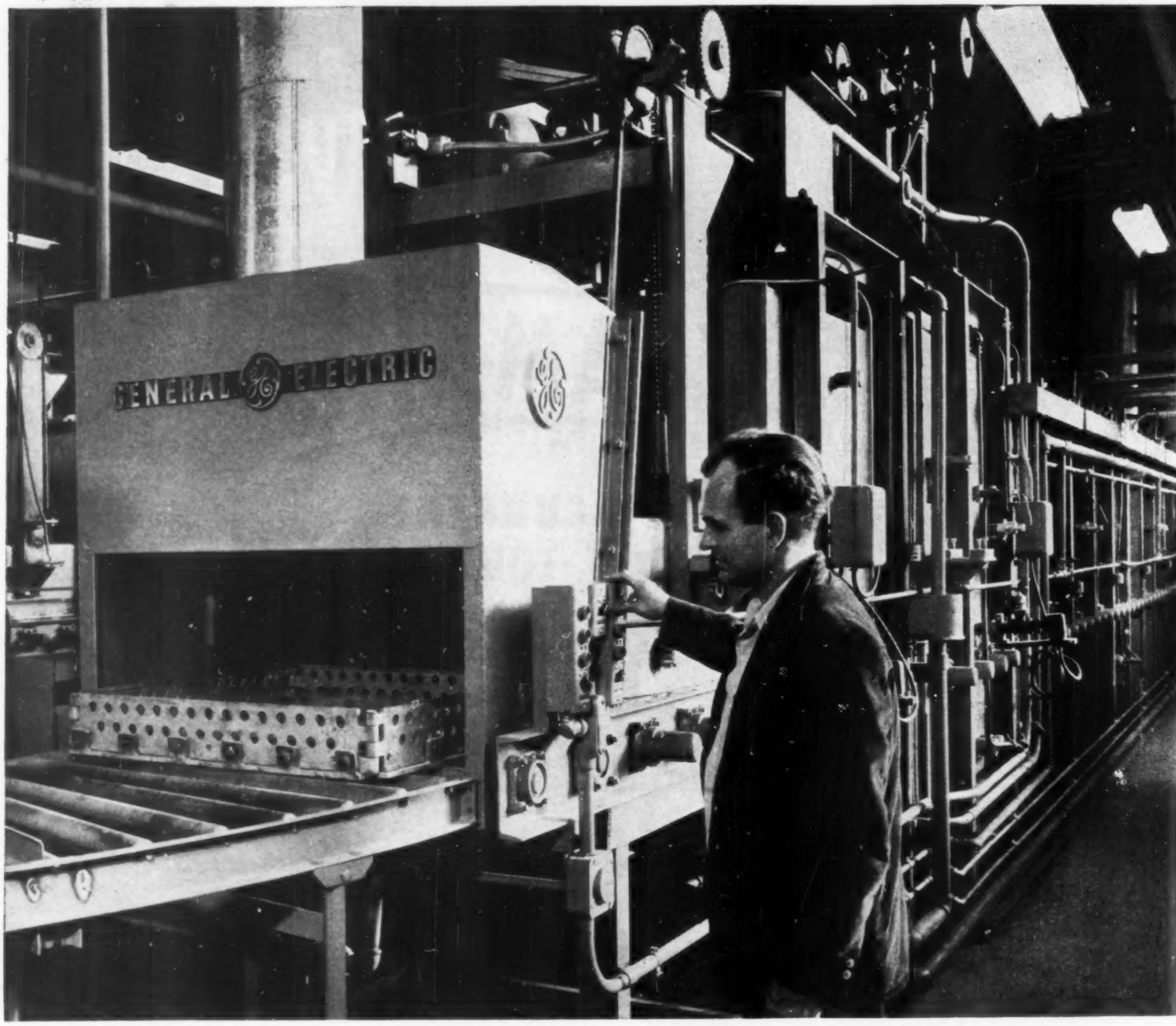
Command eye and ear attention when time interval is completed. Automatically closes or opens circuit at end of elapsed time, and operates buzzers, bells, or lights at remote stations. 5"x5"x3", readily attached to wall, panel, or switch box. 115 to 230 volts AC, 25, 50, or 60 cycles. Slow-speed, self-starting motor. Pure silver contacts. Interval range, 1 min. to 3 hrs., dial calibrated, 1 sec. to 5 min. *Complete information in Bulletin #98*

Manufacturers of these and other timers and controls for Industry:
AUTOMATIC RE-CYCLING TIMERS • TIME-DELAY TIMERS • RUNNING TIME METERS • INTERVAL TIMERS.

Timers that Control the Pulse Beat of Industry

INDUSTRIAL TIMER CORPORATION
131 OGDEN STREET, NEWARK 4, N. J.

LINK-BELT ball and



Jet engine parts are heat-treated in General Electric roller-hearth furnaces at the Aviation Division of The Bingham-Herbrand Corporation, Freemont, Ohio. To counteract high temperatures, the Link-Belt ball bearing flanged blocks are water-cooled. Like other manufacturers of equipment that must operate efficiently under the toughest service conditions, GE utilizes various types of precision Link-Belt bearings.

LINK-BELT ball and roller bearings

Metal Working Machinery

Farm Machinery

Construction Equipment

Textile Machines

roller bearings

...keep company with America's hardest-working machines

WHERE loads are heaviest and operating conditions most severe—you can always rely on Link-Belt precision ball and roller bearings. That's why so many top equipment designers specify Link-Belt for their tough-job applications.

Thousands of installations have proved the dependability of Link-Belt bearings. Quality construction throughout—combined with precision manufacture—assures the ultimate in accuracy and long life.

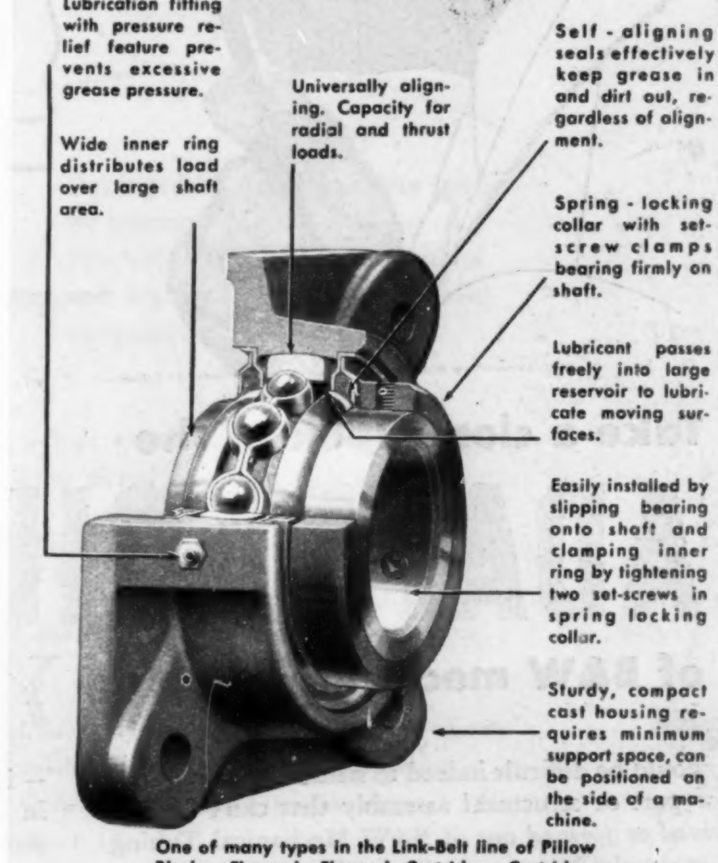
There are Link-Belt ball or roller bearings to meet every industrial need. And there's a Link-Belt bearing engineer near you to help apply them to your particular requirements. You can also get all the facts from Data Book 2550.

LINK-BELT

BALL and ROLLER BEARINGS

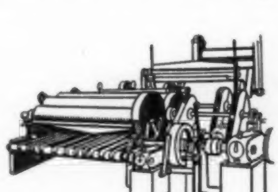
LINK-BELT COMPANY: Executive Offices, 307 N. Michigan Ave., Chicago 1. To Serve Industry There Are Link-Belt Plants, Sales Offices, Stock Carrying Factory Branch Stores and Distributors in All Principal Cities. Export Office: New York 7; Canada, Scarboro (Toronto 13); Australia, Sydney; South Africa, Springs. Representatives Throughout the World. 13,273-A

Cutaway of Series 200 Ball Bearing Flanged Block

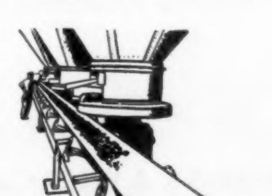


One of many types in the Link-Belt line of Pillow Blocks, Flanged, Flanged Cartridge, Cartridge, Hanger, Take-Up Blocks and Unmounted Bearings.

smooth the path of power for all industry

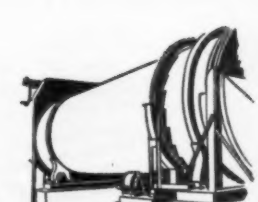


Pulp & Paper Machinery

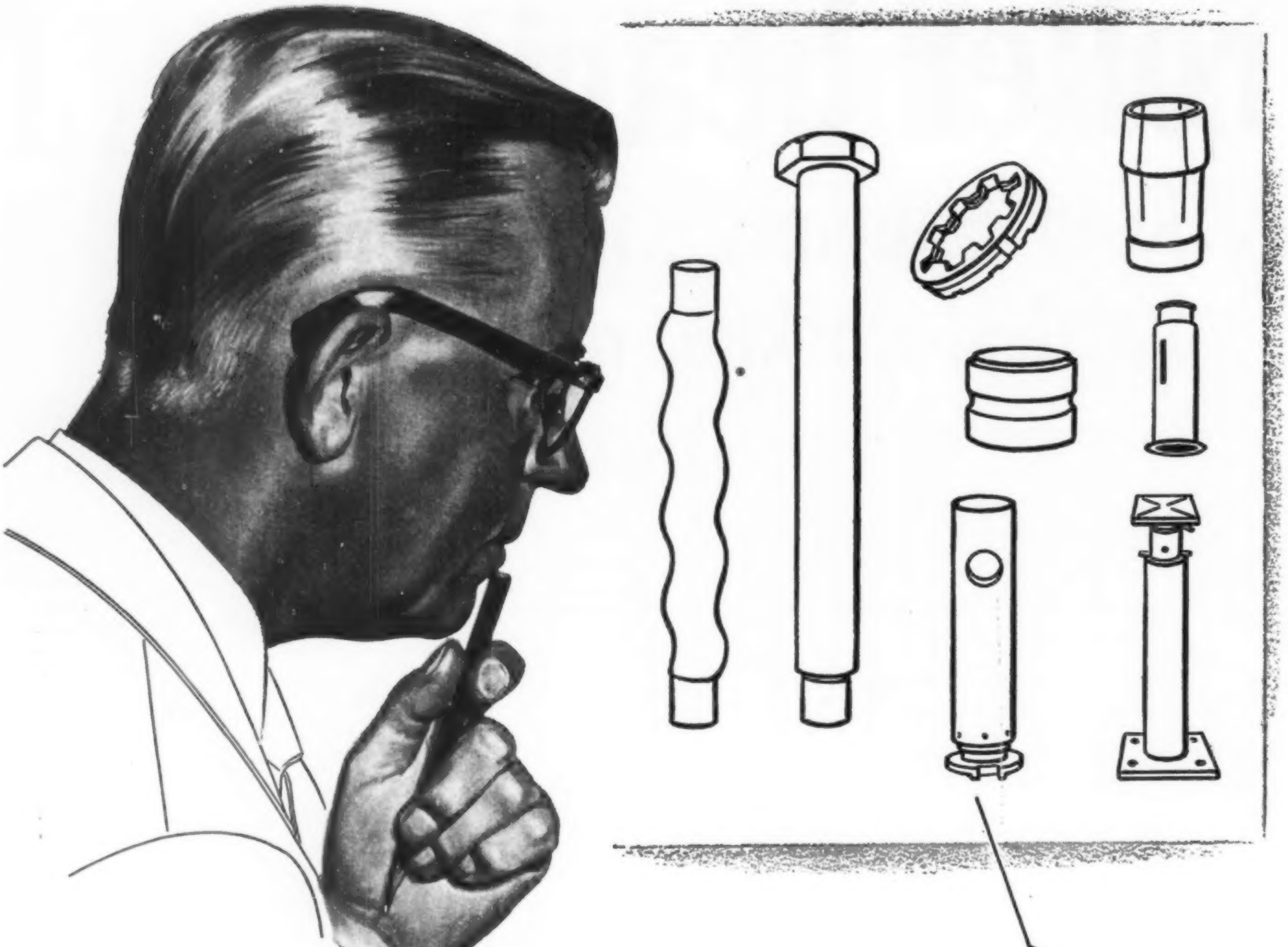


Materials Handling Equipment

Oil Field Equipment



Processing Equipment



Take a closer look at the

ADAPTABILITY

of B&W mechanical tubing *You'll like it*

It would be difficult indeed to name any round hollow part or structural assembly that can't be *machined* or *formed* out of B&W Mechanical Tubing – at worthwhile economies in time, costs and materials. It will pay you, as it has so many others, to look into its great versatility and the opportunities it affords for cutting production expense and improving product quality.

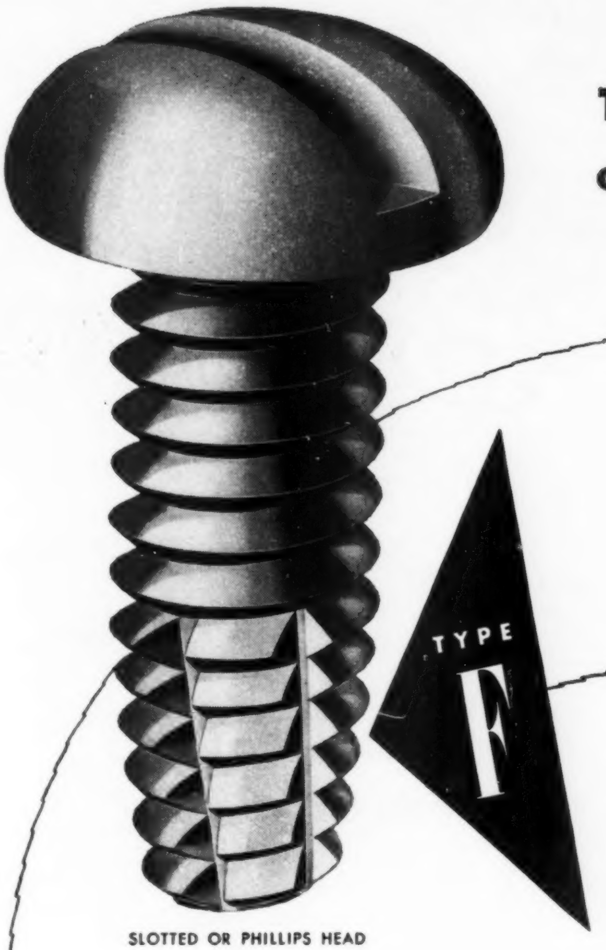
B&W Mechanical Tubing—both seamless and welded – is available in stainless, alloy and carbon steels, in tempers, grades, finishes and sizes to suit any needs.

Call on Mr. Tubes – your nearby B&W Tube Representative – if you want help in determining the mechanical tubing best suited to your needs. Get the benefit of his long, close association with mechanical applications of all kinds to help you cut costs while keeping production high.

B&W SEAMLESS and WELDED TUBES

THE BABCOCK & WILCOX COMPANY
TUBULAR PRODUCTS DIVISION
Beaver Falls, Pa.—Seamless Tubing; Welded Stainless Steel Tubing
Alliance, Ohio—Welded Carbon Steel Tubing

TA-1747(C)



SLOTTED OR PHILLIPS HEAD

The simpler way to save *
on fastenings to metal and plastics

PARKER-KALON Hardened Self-tapping Screws

Originated by P-K, first choice today for fastenings to sheet metal, structural steel, non-ferrous castings and forgings, plastics, plywood, hard compositions. Here's why:

FASTER. No tapping. Simpler, faster than machine screws, riveting, or bolting. Cuts a standard machine screw thread as it's turned in.

STRONGER. No loose fits, mistapping, or cross-threading. Lasting tightness under vibration proved in millions of applications.

LOWER COST. Saves tapping time and tool expense, permits use of power drivers for faster assembly.

IF IT'S P-K . . . IT'S O.K. For guaranteed quality, get the Original, — specify P-K. Samples and application information free. Ask your P-K Distributor or write: Parker-Kalon Division, General American Transportation Corporation, 200 Varick St., New York 14.

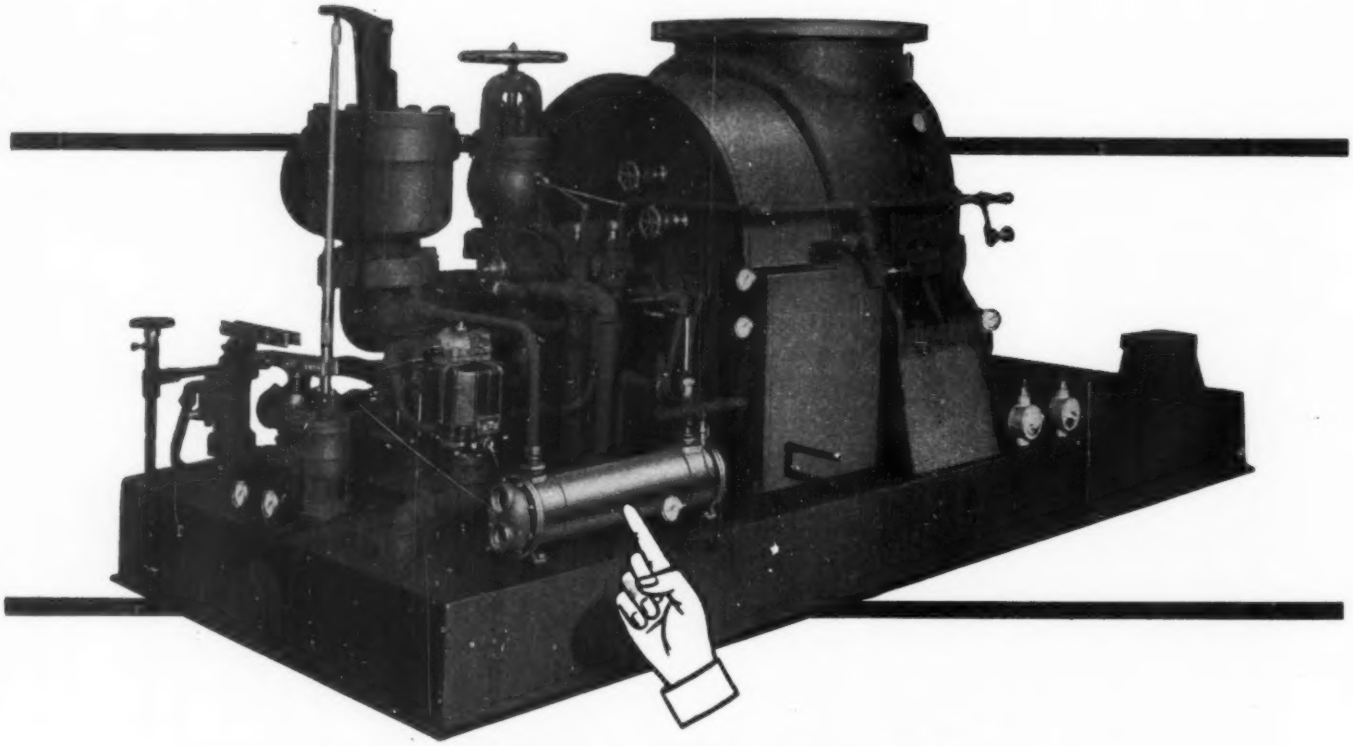
* Up to 50% and more. Ask a P-K Assembly Engineer to help you estimate your saving.

The INDUSTRIAL DISTRIBUTOR steers your Supply Dollars to the best values.

PARKER-KALON®
The Original **SELF-TAPPING SCREWS**

Feature-packed
SIZE 4
STARTER
LATEST ADDITION
Balanced
"Hook-on" base design saves installation time and money
Special sintered metal contacts last longer
High arc-interrupting capacity with "magnetic yoke" arc chamber
Coil and contacts removable from front without disturbing power wiring
Permanent air-gap lengthens magnet life
Up to 8 interlock circuits (4 N.O. and 4 N.C.) easily front-mounted
New coil holder simplifies coil change
All parts front-mounted for easy service and maintenance
SQUARE D COMPANY
1903 · 50 YEARS OF DESIGN LEADERSHIP · 1953

keeping oil temperature at proper operating level in this Terry Turbine



A ROSS EXCHANGER

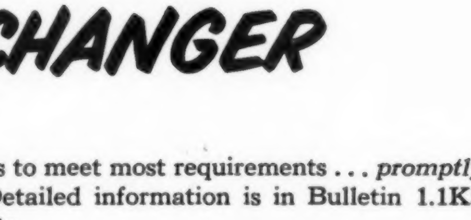
Driving a centrifugal compressor in demanding refinery service, this Terry Multi-Stage Turbine is rated 2700 hp at 8500 rpm. To provide oil temperature control for the unit's variable speed oil relay governor, main journal and thrust bearings, The Terry Steam Turbine Company factory-installed a Ross Type BCF Exchanger. An adequate supply of dependably cooled lube oil, at correct viscosity, is thus assured at all times.

Regularly written into specifications where high thermal efficiency and extreme ruggedness are of prime importance, Ross Exchangers are widely preferred throughout the oil and gas industry as standard components of engines, compressors, speed increasers, transformers and hydraulic equipment.

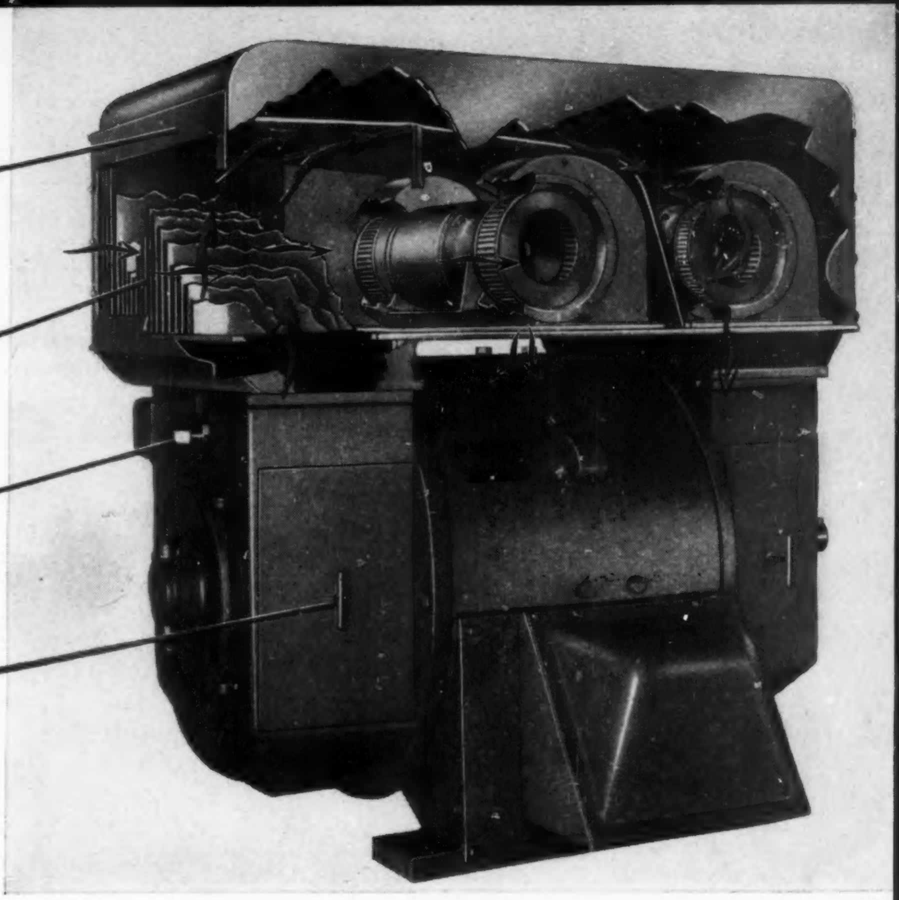
Pre-engineered, fully standardized, all copper and copper alloy Ross Type BCF Exchangers are stocked in a wide range of sizes to meet most requirements . . . *promptly.*

Detailed information is in Bulletin 1.1K5. Write.

KEWANEE-ROSS CORPORATION
DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION
1429 WEST AVENUE • BUFFALO 13, N. Y.
In Canada: Kewanee-Ross of Canada Limited, Toronto 5, Ont.



Serving home and industry: AMERICAN-STANDARD • AMERICAN BLOWER • CHURCH SEATS & WALL TILE • DETROIT CONTROLS • KEWANEE BOILERS • ROSS EXCHANGERS • SUNBEAM AIR CONDITIONERS



COMPACT, top-mounted cooling unit falls within the over-all length and width of main motor.

SMOOTH vertical plates are self-cleaning. Cooling unit is easy to remove for inspection.

PROTECTIVE THERMOSTAT helps safeguard main motor in case of blower-motor power failure.

EASY ACCESS to inspection opening—no tools required.

General Electric Announces New Unit-Cooled D-C Motor!

THE SMOOTH STYLING of the new unit-cooled motor is in harmony with good machine design. Smooth contours facilitate cleaning. No ducts or piping are needed on this compact, ready-to-install unit.

Smallest of type from 15-200 hp, new motor features 50% greater heat transfer in 37% smaller unit than previous design.

To meet your industry's need for a compact, totally enclosed motor, this new General Electric unit-cooled design features more effective heat transfer in a smaller unit.

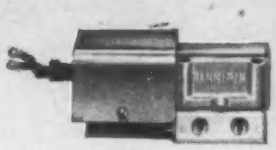
Special construction assures dependable operation, with minimum maintenance, in the severe atmospheres which make open or partially enclosed motors difficult to maintain.

Shutdowns for cleaning and inspection are reduced to a minimum. Air-borne contaminants and foreign particles are kept out of the internal ventilating air by the totally enclosed construction

For additional information on ratings and dimensions, contact your nearest General Electric office or write for Bulletin GEA-6091, "Unit-cooled D-C Motors." Address Section 770-31, General Electric Company, Schenectady 5, N. Y.

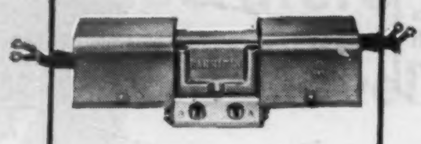
You can put your confidence in—

GENERAL ELECTRIC



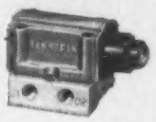
CC1-25, Solenoid-Operated Spring Return

CC6-25, Foot Operated

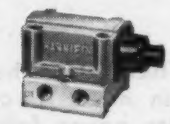


CC11-25, Solenoid-Operated Momentary Contact Type

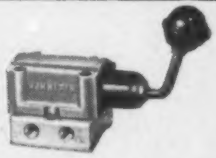
CC7-25, Foot-Operated Locking



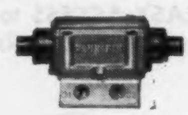
CC2-25, Ball Cam-Operated



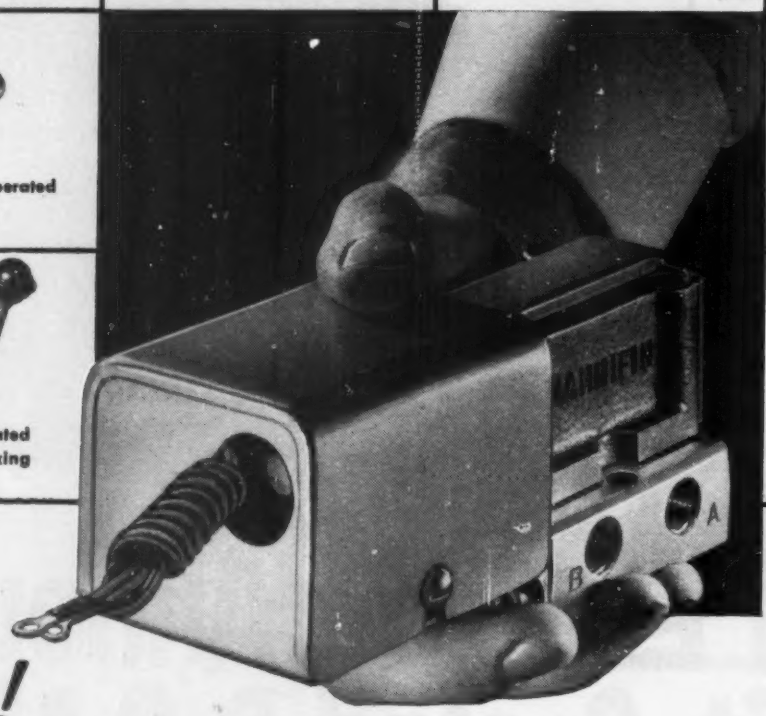
CC10-25, Piston-Operated Spring Return



CC4-25, Hand-Operated
CC5-25, (Same), Locking



CC12-25, Piston-Operated Momentary Contact Type



Now!

¼" P-M 4-Way Valves!

more compact, less expensive than any similar valves

Here's a new series of direct-operated 4-Way Valves for the control of small, double-acting air cylinders. It's the latest addition to Hannifin's revolutionary "P-M" line.

Designed for easier maintenance. The only moving part is the valve stem with its two poppets. The valve body, which contains this stem, is quickly removed from the separate "manifold" to which all pipe connections are made. Thus, the entire valve can be serviced without breaking line connections.

The CC Series. These new "P-M" 4-Way Valves, nominally of ¼" pipe size, flow so much air that they can be used competitively with other ⅜" valves when ordered with ⅜" ports. They are corrosion resistant throughout and meet J.I.C. recommendations.

Compare! You'll find these new Hannifin 4-Way Valves more compact, simpler and easier to use, and *less expensive* than any comparable valves.

GET THE COMPLETE HANNIFIN "P-M" AIR CONTROL VALVE CATALOG

The new Hannifin Controls Catalog contains complete information and specifications on all Hannifin "P-M" Valves, including this new 4-Way direct-operated series. Write for Bulletin 232.

HANNIFIN

Hannifin Corporation, 1115 S. Kilbourn Ave., Chicago 24, Ill.

Air and Hydraulic Cylinders • Hydraulic Power Units • Pneumatic and Hydraulic Presses • Air Control Valves

Aluminum, like no other metal, combines good looks, light weight, high heat and electrical conductivity, corrosion and red-rust-resistant properties. What's more, its excellent machinability spells fast, accurate, low-cost production of intricate shapes by hand screw machine and single and multiple-spindle automatic screw machines. When you consider that three times as many parts can be machined from a pound of aluminum as from a pound of heavy metal, you find its cost less than for most other metals.

For these reasons, needle valves, electrical switch components, gears and camera parts are fabricated from aluminum as well as all types of fastenings—wood screws, machine screws, bolts, cap screws and nuts.

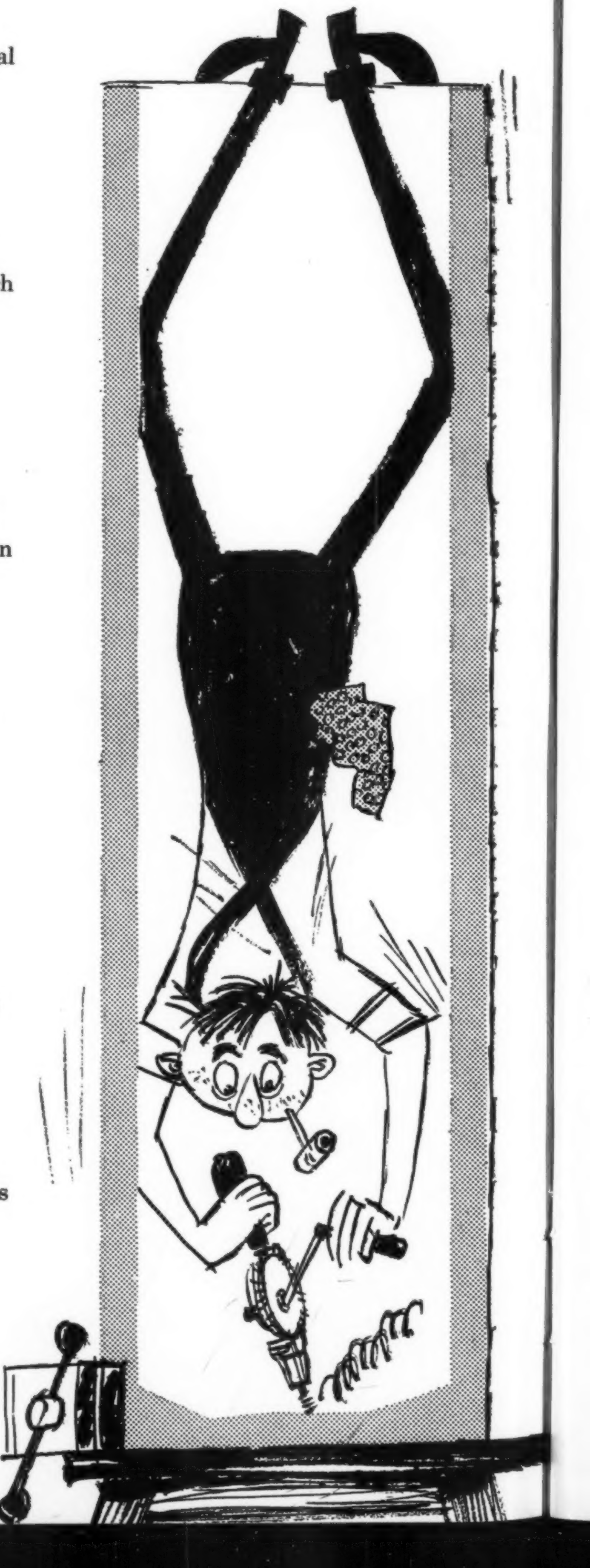
Aluminum permits faster feeds and speeds, but each alloy is a different metal, and machining characteristics of each must be taken into account. Alcoa, with years of experience machining aluminum, has the background necessary to make proper alloy selection for a given product. Too, it takes great care in tool choice, condition and setting—the keys to accuracy and good finish.

Someday, you may ask yourself, "Why aren't we using aluminum screw machine products?" To help you provide the answer, we submit that the following is . . .

What tomorrow's Design Supervisors should know about Aluminum Screw Machine Products and Fasteners today

About Alloy Selection

"... deep drilling where hole depth is greater than four to six diameters"



In choosing an alloy, it is important to select the easiest to machine—that in addition possesses all the necessary physical and mechanical properties your product demands.

Alloy 11S-T3 is the best all-around alloy for machining. It is used in a wide variety of general applications where good machinability is important. Its free-machining elements, bismuth and lead, account for this and also the fact that good finishes can be achieved even at high speeds. In fact, the only limiting factor today to the speed at which 11S-T3 can be turned is the speed of the machine itself.

Alloy 11S in the T8 condition is artificially aged 11S-T3, and has greater strength than the latter with similar machining characteristics. It is the better of the two for deep drilling applications where hole depth is greater than four to six diameters.

Of the heat-treatable alloys, 17S-T4, 24S-T4 and 61S-T4, 17S-T4 is most commonly used for its high strength or when deep drilling is called for. Alloy 17S in the T4 condition should be employed for general uses because of its higher mechanical properties and lower material cost.

Alloys 24S-T4 and 61S-T4 are the next most commonly used of the heat-treatable alloys. When product application calls for high resistance to corrosion, where anodic finishes are to be applied or where parts are to be machined from tubing, alloy 61S-T4 is used. Alloy 24S-T4 is now widely accepted as the nut, bolt, screw and washer alloy because it has higher strength. Too, there is an advantage in standardizing on one aluminum alloy for this type of use.

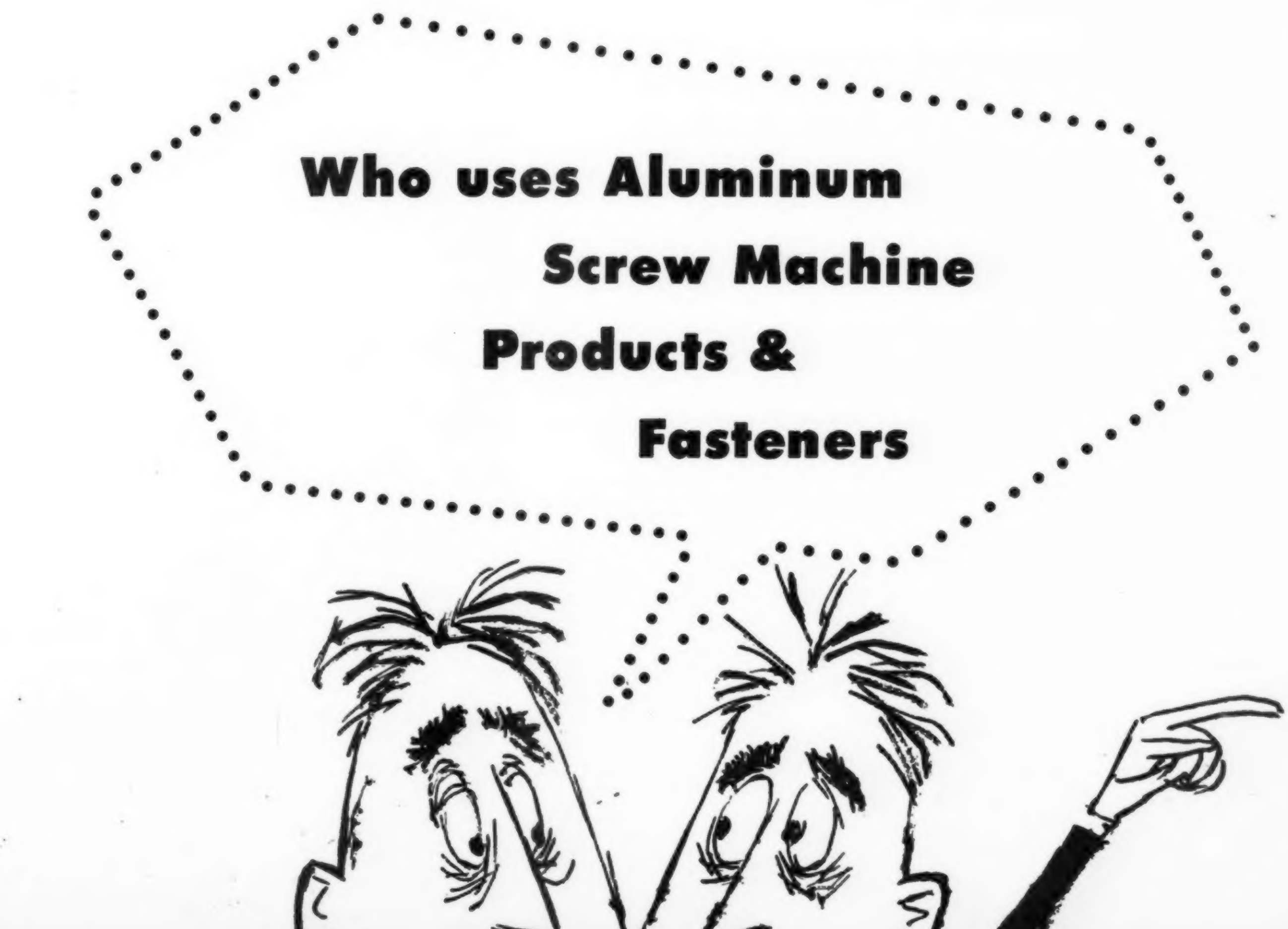
With alloys containing 98.7 and 98.8 per cent aluminum, there is higher electrical conductivity with a slight sacrifice in machinability. Alloy 2S, which is slightly harder than the purer alloys, gives better machinability and is preferred over the purer alloys because of its higher strength. It is especially good when severe forming operations, such as spinning ends closed, flaring, flattening or upsetting, are to follow machining. Its high resistance to corrosion is used to good advantage in fittings for cooking utensils, for applications that require welding, conduit clamps, for gasket and sealing washers where close fit demands good deformation.

About Physical Properties

The aircraft industry was one of the first to recognize the advantages of aluminum in any fabricated form—largely because of its good strength-weight ratio. And that holds true for aluminum screw machine products and fasteners as well. For example, 24S-T with its light weight, 0.100 pound per cubic inch, has a yield strength of 48,000 psi and a tensile strength of 68,000 psi.

There is an advantage in machining battery caps and molded plastics inserts of aluminum. Since the coefficient of expansion of aluminum is similar to that of most plastics, the often severe disadvantage of combining materials with dissimilar rates of expansion and contraction no longer presents a problem.

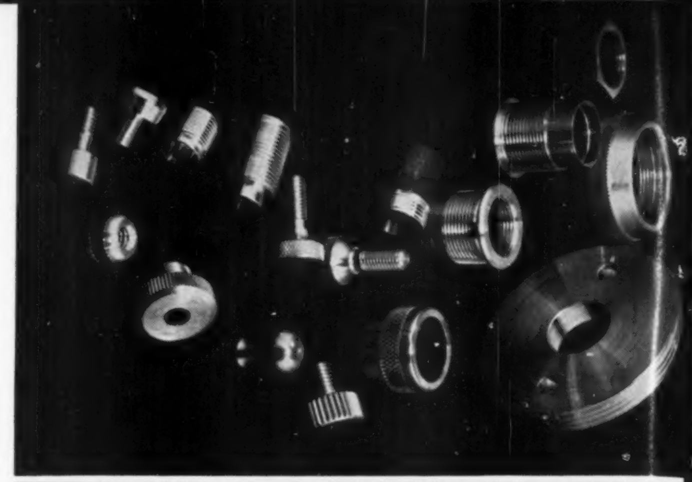
Although cutting speeds do vary according to the alloys and part size, surface speeds up to 700 sfm are standard for turning round bar stock with a box tool. With square or hex stock, surface speeds of 300 to 450 sfm are common. And recommended feeds are equal to or above those used for most other metals depending on the depth of the cut, the tolerance and surface finish required. Generally, tolerances are quite close, but may be improved by slowing the feed. Surface finishes usually run better than five micro-inches with a box tool.



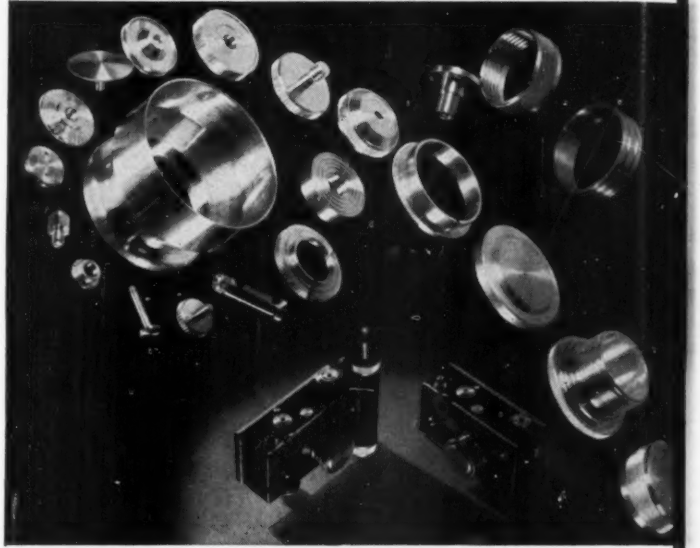
Who Uses Aluminum Screw Machine Products & Fasteners

Probably more parts of all sizes and shapes are produced by machining than by any other primary fabricating process. True, during the war, most of Alcoa's output of screw machine products went directly into the war effort.

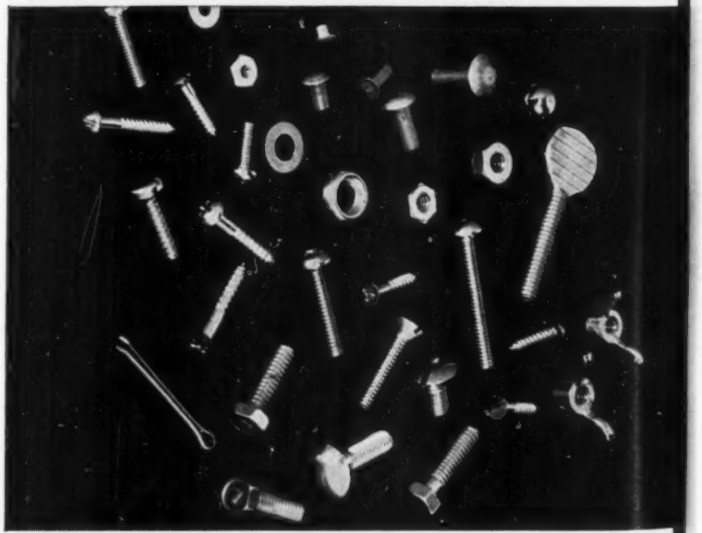
But, today, in addition to supplying defense needs, machined parts are going into a staggering variety of products. Listed here are a few. Is yours among them?



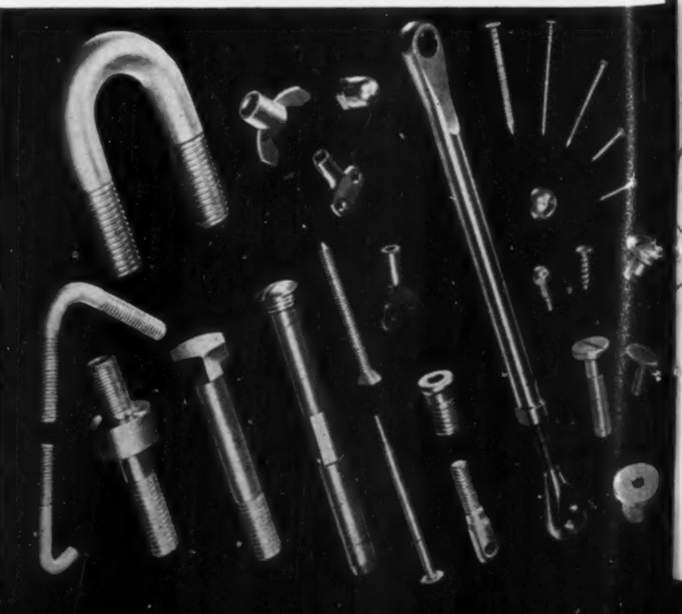
ELECTRICAL SWITCH COMPONENTS—Aluminum's high electrical conductivity makes it a natural for use in electrical switch components. And, with its high resistance to corrosion and chemical attack, it is invaluable when exposed to weather or in use about process industry areas. It is lightweight, clean and bright. Best of all, despite its slightly higher raw-material cost, its end cost is lower since three times the number of electrical components can be turned from a pound of aluminum as from a pound of heavier metal.



CAMERA PARTS AND ACCESSORIES—Aluminum is rapidly replacing other metals in cameras of all kinds. Its naturally good-looking finish, especially when enhanced by special anodic or chromium finishes, imparts long-lasting good looks and extra "buy appeal" to any product. Machined from 11S and 17S alloys, these camera parts are light, strong and will give years of dependable service.



STANDARD FASTENERS—In assembling aluminum parts, it is best to use aluminum fasteners, first, to avoid any galvanic action, and second, to provide a uniform appearance. Thousands of standard fasteners are kept in stock at Alcoa to provide prompt service to your orders. Among these are machine screws, wood screws, sheet-metal screws, bolts, nuts and washers.



SPECIAL FASTENERS—Where product design specifies and quantity justifies fasteners of special design, Alcoa's complete hand and automatic screw machine facilities as well as cold-heading equipment stand ready to produce them to your specification. Such fasteners are available in a wide range of alloys, types, sizes and finishes. Added protection of an Alumilite* finish can be furnished if desired.

*Trade Name of Aluminum Company of America

About Finishes for Aluminum Screw Machine Products and Fasteners

Choose a finish for your product that is in keeping with its function. Ask yourself, "Does it need an especially attractive luster for attention-getting sales appeal, or would it be enough to supply the very minimum finish and rely only on aluminum's inherent lasting qualities?"

Here, finishes for aluminum are broken down into three basic types, mechanical, chemical and electrochemical, and are listed according to increasing cost.

Mechanical

Simplest of all finishes is that which is performed by the tools in shaping or machining the part. Unmachined sections have a natural oxide coating; machined sections take on the finish left by the tool. It should be remembered that the finish in heat-treated parts is characteristically dark.

When appearance is not important, a tumbled finish can be used to remove burrs or sharp edges. Tumbling can be employed prior to threading the piece. But, it should never be specified where threads or close tolerances must be preserved.

Most durable of all mechanical finishes is the "bright finish" or ball burnishing method where parts are placed in a tumbling barrel along with steel balls or cones. This type of finishing process cannot be used on parts with thin edges or where precision thread tolerances must be maintained.

Tubes, bushings and similar parts having straight cylindrical outside diameters can be furnished with a centerless grind finish. Small parts can be shipped in bulk; large parts require special packaging to avoid rubbing in shipment.

"... attractive luster for attention-getting sales appeal"

Scratchbrush, or satin finish, a modification of the scratchbrush method, is most generally used on pots and pans, but may also be used on small screw machine parts used to assemble utensils in order to give uniform texture.

Polishing, an operation which uses a fine-grit emery held in suspension by a tallow, oil or beeswax, gives what is called an "oiled" finish. It is often used just prior to final buffing, but can be used to achieve texture similar to that of the satin finish.

The buffed finish brings out a high luster, with all minor surface scratches or lines removed. Or, a second buffing operation may be used. This, the highest grade finish available for screw machine products, is performed only after preparatory grinding, oiling and first buffing have been performed. Because its extreme high polish will show marks readily, this finishing technique is rarely used unless it is to be followed by an anodizing process.

Chemical

Chemical etch is an economical way of imparting a uniform frosted finish to aluminum. Actually, a microscopically rough finish is produced which tends to hide surface scratches and give a light-diffusing texture to the surface.

Electro chemical

Alumilite or anodic coatings, when applied to screw machine products, provide finishes which are protective, decorative and functional. Threaded parts have less tendency to bind. And Alumilite offers protection against weather, wear and chemical attack. No matter the texture of the metal on which they're applied, that texture will be reproduced after the anodic coating has formed. Clear, translucent or opaque finishes are possible, but for uniformity between related parts, each should be made of the same alloy and temper.

The Alumilite finish when it is first formed is filled with smaller-than-microscopic pores. When it is important to preserve the appearance of the coating, special sealants fill those pores, making the surface stain resistant. If protection is paramount, corrosion inhibitors such as chromates may be absorbed and sealed within the coating to prepare a base for enameling. Or, dyes can be absorbed into the solution which lend a unique colored finish with all the luster of the metal showing through.

"... fine threads have a tendency to seize"

What you should know about designing for Aluminum Screw Machine Products

Although most designers are familiar with good automatic machine practice, here are a few points to keep in mind when designing for aluminum screw machine products:

TOLERANCES—Should always be as large as possible with close tolerances specified only on those dimensions critical to the part's function. Costs are directly proportional to tolerance requirements. Location of cross holes, slots, milled sections and trademarks should not be held to close tolerances unless they affect the functioning of the part. If an outside portion of the product need not be machined, that point should be designated, and standard bar stock dimensions should be given.

DRILLING—Whenever possible, specify standard drill sizes. In some cases flatter angles can be used, but absolutely flat-bottomed holes usually mean extra expense. Holes deeper than six times their diameter are difficult and expensive to drill. Wherever possible, diameter should be increased; or, possibly, two holes, each with successively larger diameters, could be drilled.

THREADS—Generally, fine threads or those having a "V" shaped contour should be avoided because of their tendency to seize. And, you should avoid threading close to a shoulder or tapping close to the bottom of a hole. Tool breakage and slower production often result.

BURRING—Where burrs occur, specify the removal of only those that must be removed. In some cases, deburring can cost as much as the machining operation itself. Closely related to the problem of burrs is that of sharp corners. Corners resulting from the intersection of two turned surfaces can often be chamfered at no extra cost. But, at the intersection of a turned and unturned section, chamfering usually requires an extra, costly operation.

SYNTHANE — making bigger payloads pay off

Synthane bushings, spacers, and bearings in the landing gear of this giant of the skies share the landing shock loads of twenty-five tons. But Synthane parts have many virtues in addition to their ability to withstand the jolts of heavy landings.

Parts made of this hard-working laminated plastic are unaffected by oils, and are dimensionally stable. They resist abrasion, and weigh half as much as aluminum.

Because Synthane is so easy to machine, it is appropriate for fair-leads and cable-sheaves in control systems. Because it is an excellent electrical insulator, you will find it at work in engine ignition systems, flight instruments, automatic controls, and radar sets. Because Synthane is light and corrosion-resistant, it's used for the flapper valves in fuel cell baffles.

Synthane has all these properties and many more. It might be a good material for you to try. Start by sending for the complete Synthane Catalog. Synthane Corporation, 7 River Road, Oaks, Penna.

IS YOUR ANSWER HERE?

If you are not in the aircraft industry, Synthane's combination of properties may still stir your interest. Besides the properties at the left, Synthane has good tensile, compressive, flexural, impact and shear strength, a low coefficient of expansion, is moisture- and wear-resistant, easy to machine.

Sliding component for handrail of moving stairway, machined from Synthane.

Breaker arms in distributor contain molded-laminated Synthane.

Synthane-one of industry's unseen essentials

SYNTHANE

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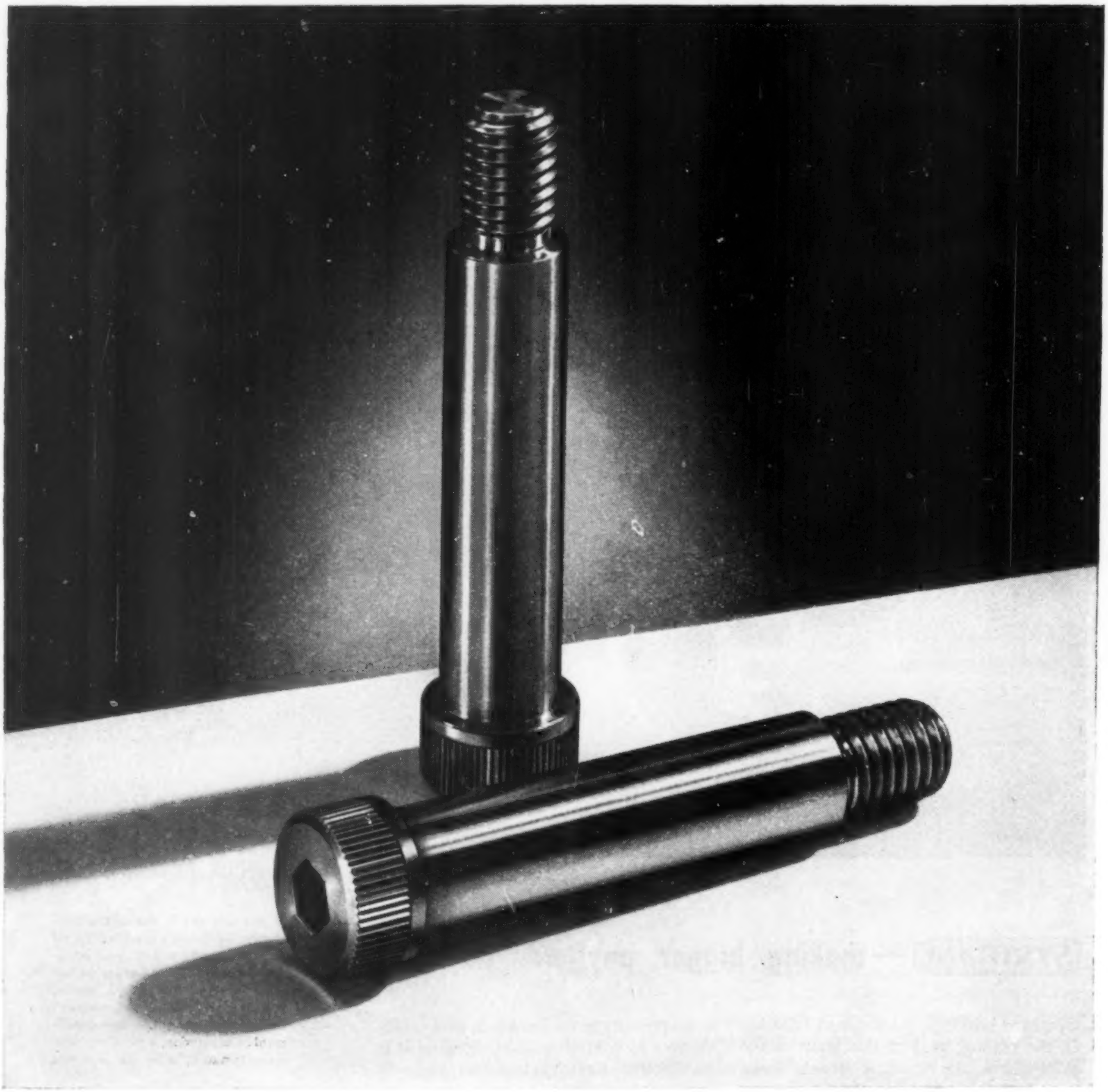
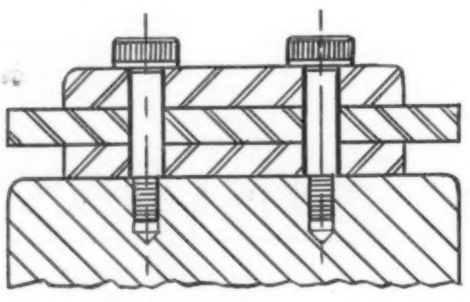
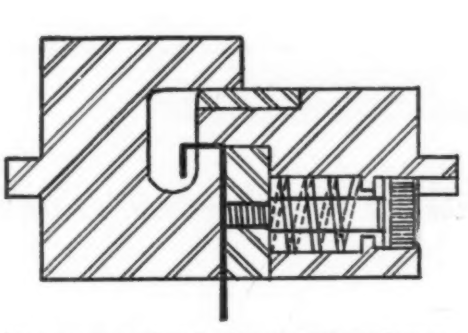
LAMINATED PLASTICS



UNBRAKO SHOULDER SCREWS have these features: heat treated alloy steel for strength; knurled head for sure finger grip, fast assembly; accurate hex socket for positive internal wrenching; shoulders held to unusually close tolerances; threads and head concentric with body for uniformly accurate assembly; finished threads close to shoulder; fully formed threads, Class 3 fit; standard sizes, ¼" to ¾" in a full range of lengths.



As stationary guides, moving shafts or pivots.



For pressure pad and stripper plate applications.

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RELIANCE V★S DRIVE® *with*

the new *Totally-Enclosed Dual-Cooled* motor

PATENT APPLIED FOR

For continuous operation over extremely wide speed ranges . . . or wherever unusually severe operating conditions must be met . . . the new Reliance Totally-Enclosed Dual-Cooled Motor extends the application range of the V★S Drive. Wherever high ambient temperatures, intermittent operation, prolonged low-speed operation, or severe duty cycles pose difficult cooling problems, Reliance V★S Drive with Dual-Cooled Motors have proved to be the best answer. The new Dual-Cooled Motor is available either totally-enclosed or explosion-proof, conforming to Bureau of Mines and Underwriters' specifications, in ratings from 15 through 150 hp. D-1404

GET THESE FACTS NOW!

Your request for Bulletin C-2201 will bring you this new booklet featuring large cutaway drawing with 3-color transparent overlay showing operation of Reliance Dual-Cooled Motor, together with fact-filled bulletin showing production increases and operating economies obtainable with Reliance V★S Drive.

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ZINC AND ALUMINUM DIE CASTINGS

The three castings illustrated are made in the same die. Provision for the changeable die parts must be included in the original die design. Expert die making is also involved, but the end results of this type of pre-planning represent substantial savings in many instances.

Madison-Kipp is skilled and seasoned in die casting mechanics and invites your inquiries for cooperative effort.

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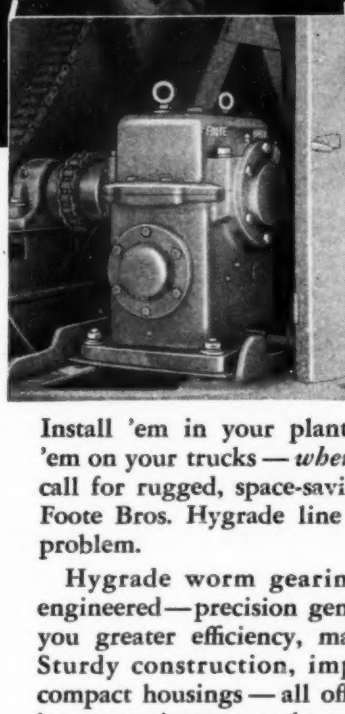

The Halliburton Oil Well Cementing Co., Duncan, Okla., keeps cement at right consistency during travel by using specially-designed Rotovoy trucks. Power is transmitted as desired through rugged Hygrade Drives.

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Install 'em in your plant — even mount 'em on your trucks — *wherever* tough jobs call for rugged, space-saving drives — the Foote Bros. Hygrade line can solve your problem.

Hygrade worm gearing is skillfully engineered — precision generated — assures you greater efficiency, maximum power. Sturdy construction, improved design, compact housings — all offer you the savings on maintenance, the superior performance you've been looking for.

Hygrade Drives are available with horizontal or vertical output shafts — vertical extending upward, downward or both. Vertical Hytop units incorporate a wider, low-speed bearing span to accommodate long, unsupported extensions. Single, helical and double-worm reductions provide you with ratios up to 4,108 to 1—capacities to 260 h.p.

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GEARMOTORS SPEED REDUCERS *with High Overhung Load Capacities*

Rynel Speed Reducers and Gearmotors are offered in a range from 1/50 to 5 H.P. and 1" to 8" centers, with any desired ratios practical in this range.

Clean-swept modern design, plus gearing built to produce exact rated efficiency makes these Rynel reduction units highly desirable for all critical operating conditions.

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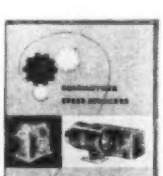

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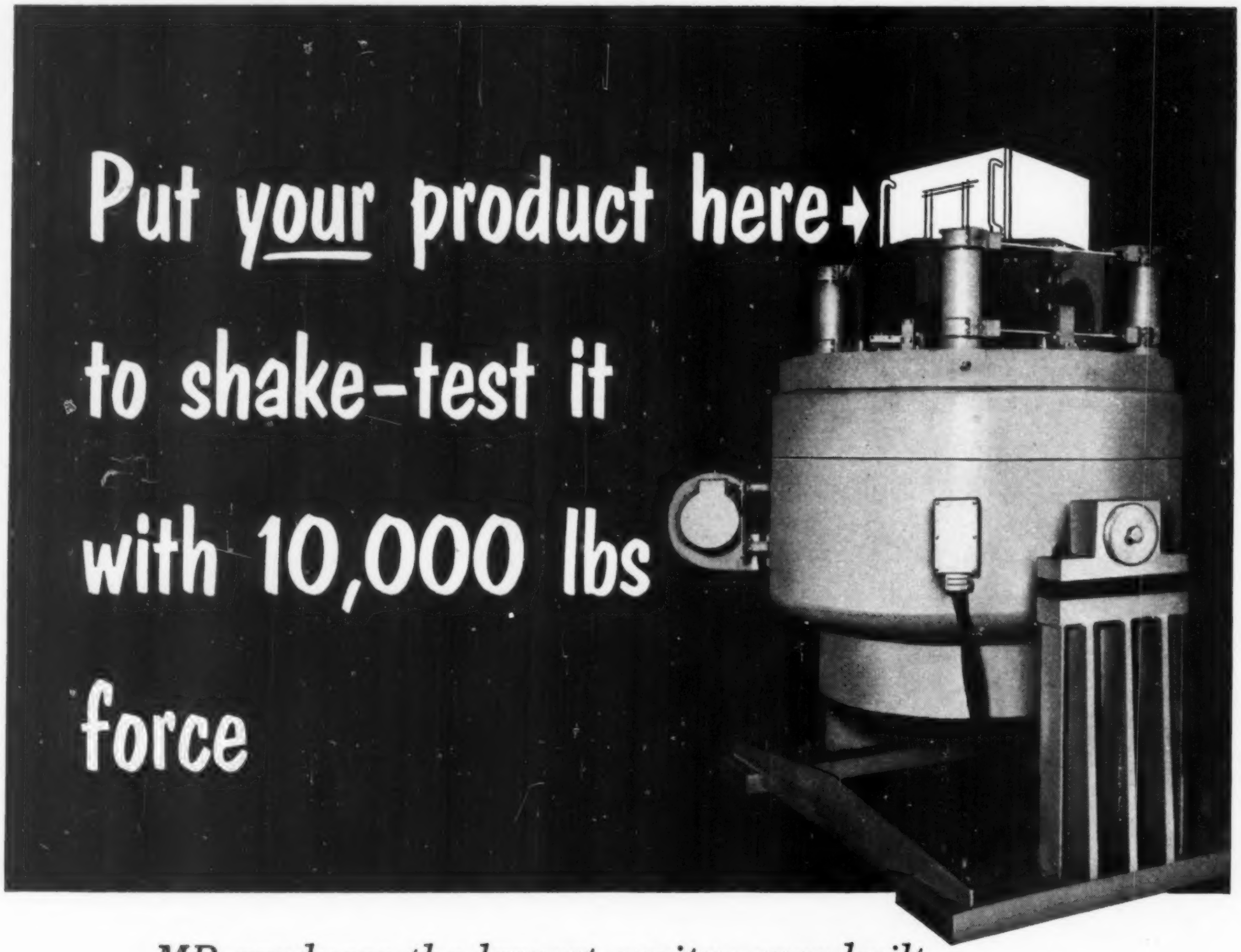

MB produces the largest exciter ever built to meet heavy duty vibration test specifications

THERE's nothing like a good shaking to test out structural designs, electronic equipment, instruments or complete assemblies for faults or flaws. In fact, for many products put to military use, such tests are *specified.* However, since *all* products encounter some vibration or shock in service, many engineering departments use an MB Exciter to test all designs. By so doing, the "bugs" are discovered in the test laboratory instead of out in the field, at cost of good will.

Largest in the line of MB electromagnetic shakers, the Model C-100 shown delivers at least 5 tons continuous force. Its performance permits heavy duty vibration testing to MIL-E-5272 and other specifications. It incorporates a number of unusual design features for easy, quick, convenient operation—including interlocking controls for complete safety and provisions for cycling tests.

HOW TO HANDLE LARGE MASSES

MB can show you a setup of vibration exciter and resonating beam that multiplies the capacity of versatile MB Exciters many fold.

Shaker being used in this fatigue strength test of aircraft engine mounts is the model S-3 rated at 200 lbs. Others available down to 10 lbs. force output.

Vibration is MB's specialty. You're invited to draw on the benefits of this specialization—and get highly qualified products for testing and control, and technical help on your problem.

Valuable bulletins for test engineers
Calibrating vibration pickups to 2000 cps is comprehensively covered in MB Bulletin No. C-11-1. Bulletin No. 1-VE-1 describes vibration exciters and details their specifications. Write today.

THE MB MANUFACTURING COMPANY, INC.
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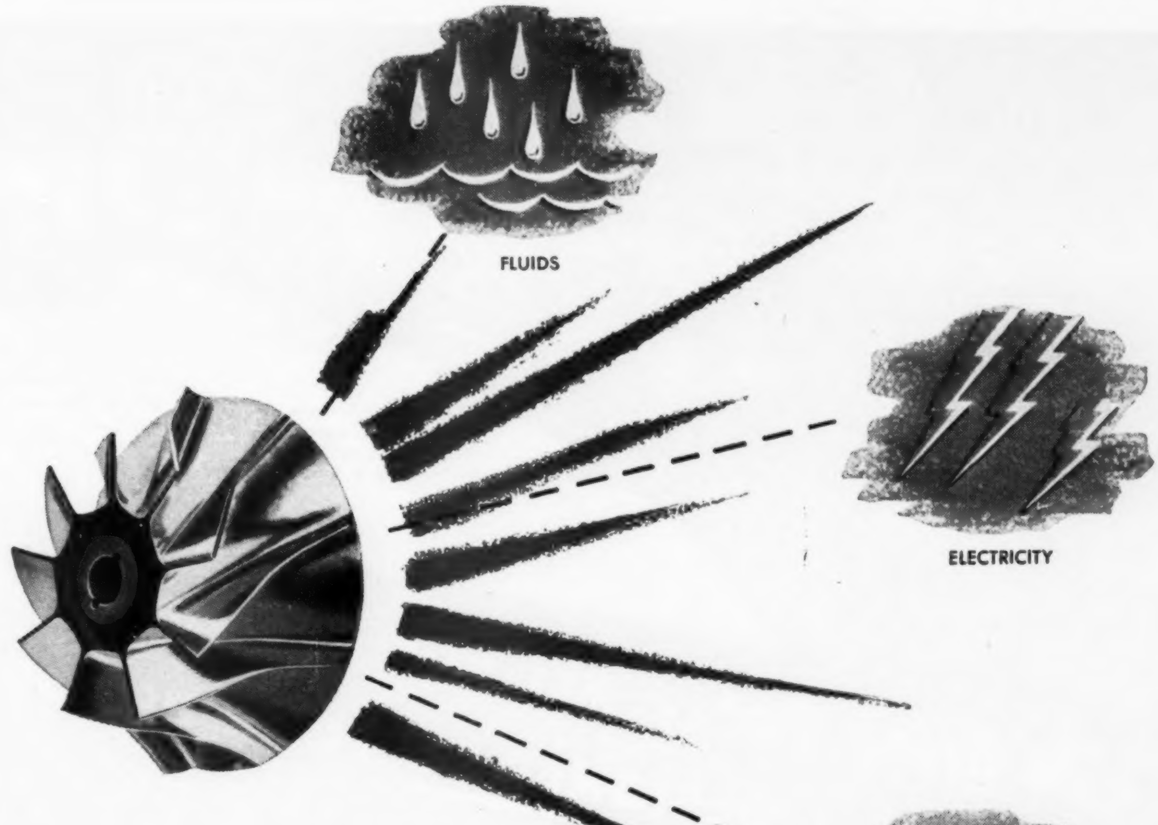

This 12 lb. harness for a 65 H.P. Cyclone ...may hold an idea YOU can use

Tucked away near the engine of a jet fighter, a Thompson Air-turbine drives an integral pump that feeds the jet after-burners.

Coupled to an electrical alternator, a Thompson Air-turbine drive delivers vast amounts of horsepower at micro-precise speed, regardless of fluctuations in load.

These Thompson turbines pack abundant power into small space, with minimum weight. What's more, they use air, always available at low cost.

Industry is discovering dozens of new uses for these Thompson Air-turbines . . . in a wide range of sizes and power. You may find it worth-while to call in a Thompson representative to tell you what we know about designing and building Air-turbines. To find out how you can put them to work in *your* application, present or future, write . . .

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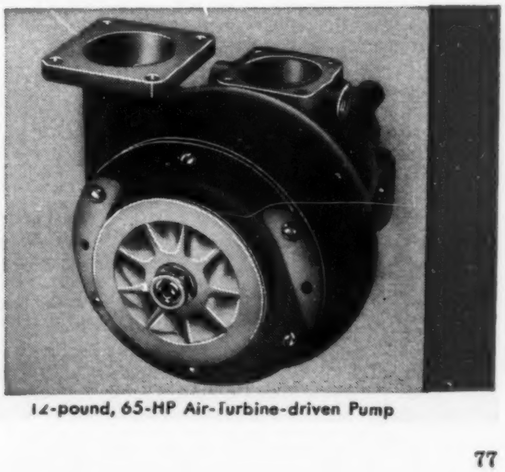

12-pound, 65-HP Air-Turbine-driven Pump


TYPE CSH—Capacitor Start, Induction, Single Phase Motor suitable where high starting torque and normal starting current is required. 1/8 to 3/4 H.P.
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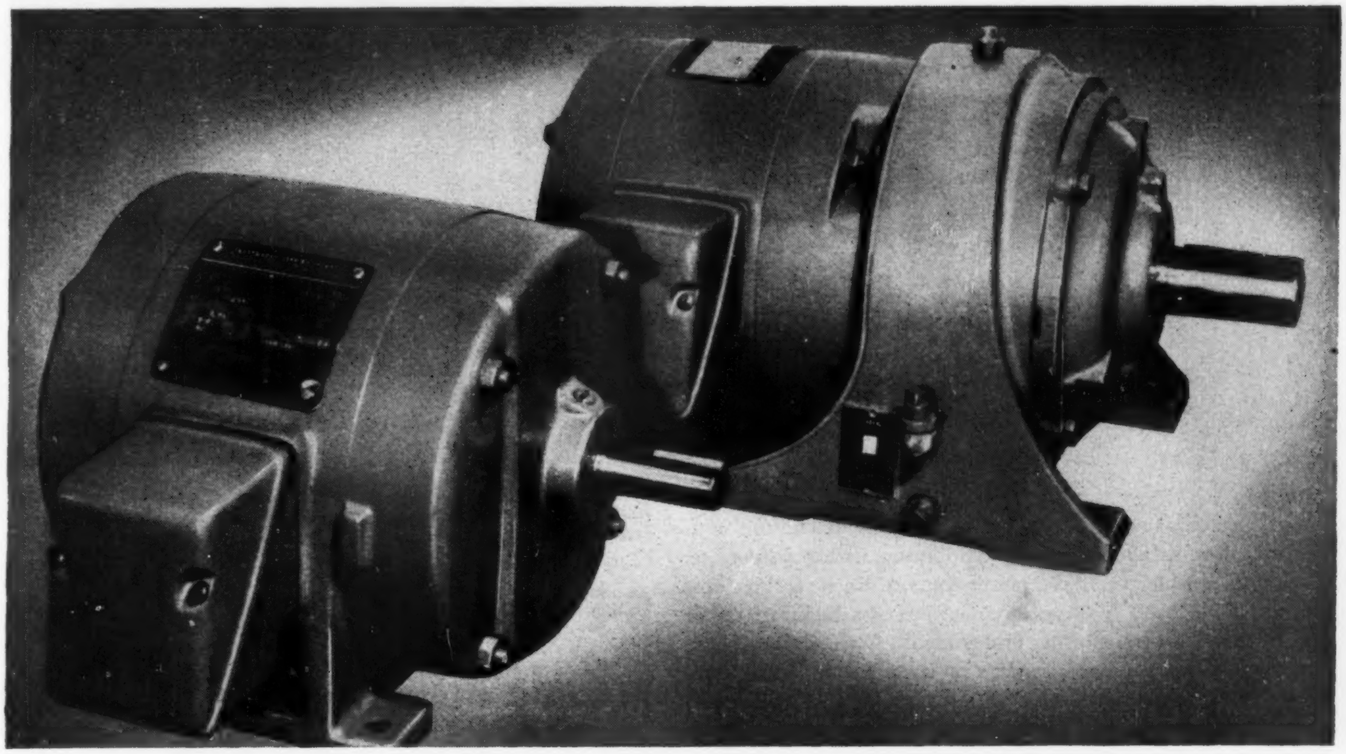
Digest

G-E MOTOR & CONTROL HIGHLIGHTS



DRIPPROOF AND GEAR-MOTORS, completely new inside and out, assure longer motor service life.

with new *TRI 55 CLAD* motors

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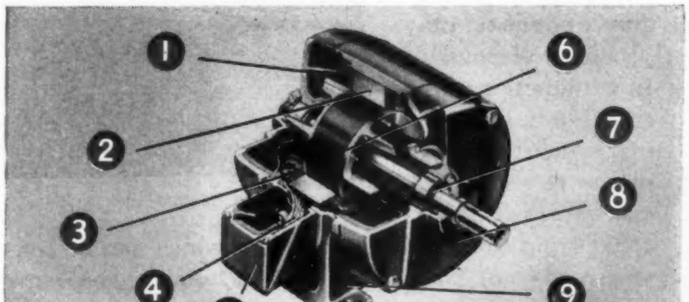

NEW MOTOR DESIGN INCLUDES: (1) polyester film slot, phase insulation; (2) press fitted stator punchings; (3) water-shedding stator windings; (4) perma-numbered leads; (5) easy-access conduit box; (6) solid-cast rotor windings; (7) advanced bearing system; (8) protected end shields; (9) cast-iron construction.

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- ☐ GEA-6012 Tri/Clad '55' Enclosed Motors
- ☐ GEA-6013 Tri/Clad '55' Dripproof Motors
- ☐ GEA-6027 Tri/Clad '55' Gear-Motors

CONSULT YOUR SWEETS PRODUCT DESIGN FILE. You'll find "everything electric" for machinery manufacturers in the General Electric section.

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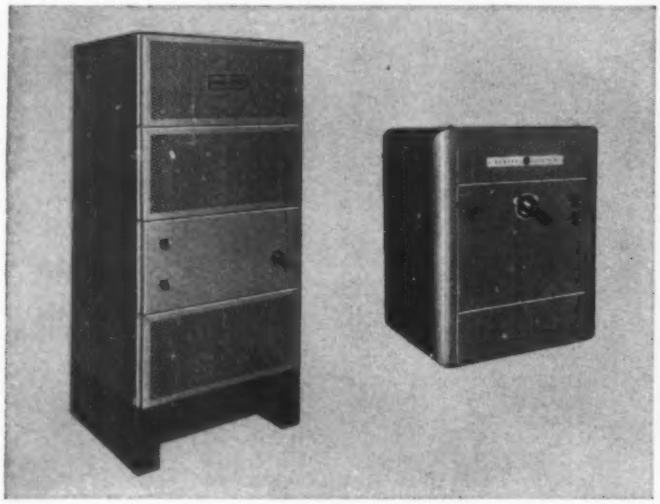
TURN PAGE FOR MORE G-E PRODUCT HIGHLIGHTS

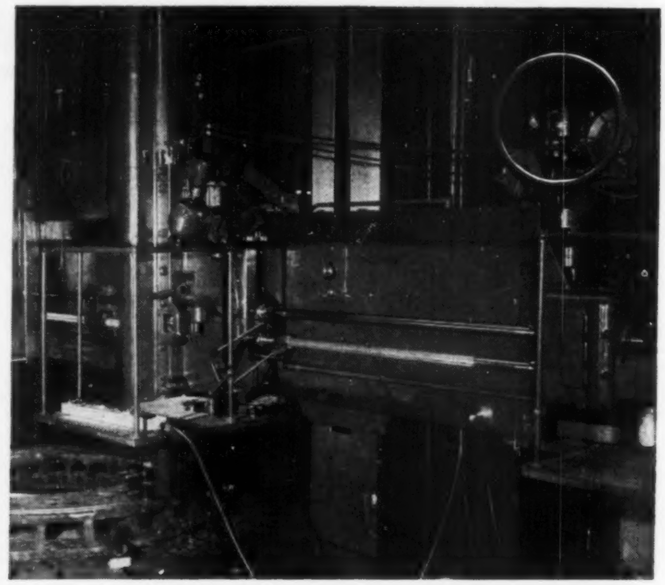
PACKAGED METALLIC RECTIFIER UNITS SATISFY MANY DC MACHINE REQUIREMENTS



Specify a compact G-E metallic rectifier for maximum convenience, low installation cost, and minimum user maintenance, when you consider using d-c power sources. Each rectifier unit is a metal-enclosed "package," comprised of selenium rectifier stacks, insulated transformer and controls. Compact in design, they save valuable space, require no special foundation, offer quick and simple installation. For example, the 25-kw unit shown above on the left is only 25 x 31 x 65⅛ inches. Ratings range from 0.75- to 25-kw (125 and 250 volts d-c) for single units and up to 300-kw for complete installations. Check Bulletin GEA-5658.

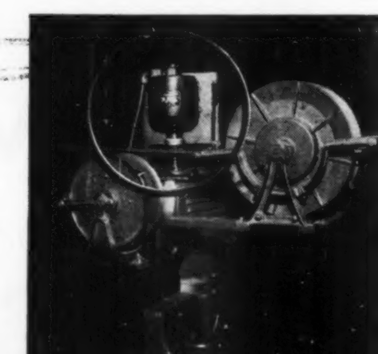
FULL LINE OF PHOTOELECTRIC RELAYS MEETS WIDE APPLICATION, LASTS LONGER

Versatile G-E photoelectric relays are easy to install, adjust, and maintain. They provide an inexpensive and accurate way to count, control, or signal. Sturdy construction gives longer service life, easier installation. Various enclosures for specific applications are available in the line. Relay shown here assures 450 operations per minute. Other models from a complete line of relays, ranging from simple to the highly sensitive units, cover up to 600 operations per minute. For information on the entire line check Bulletins GEA-3533, GEA-5920, and GEA-5921.



PRECISE SPEED CONTROL on this large boring mill is provided by a G-E 1000-rpm, 50-volt tachometer-generator connected to the feed drive—assures optimum milling performance, smooth surface finishes.

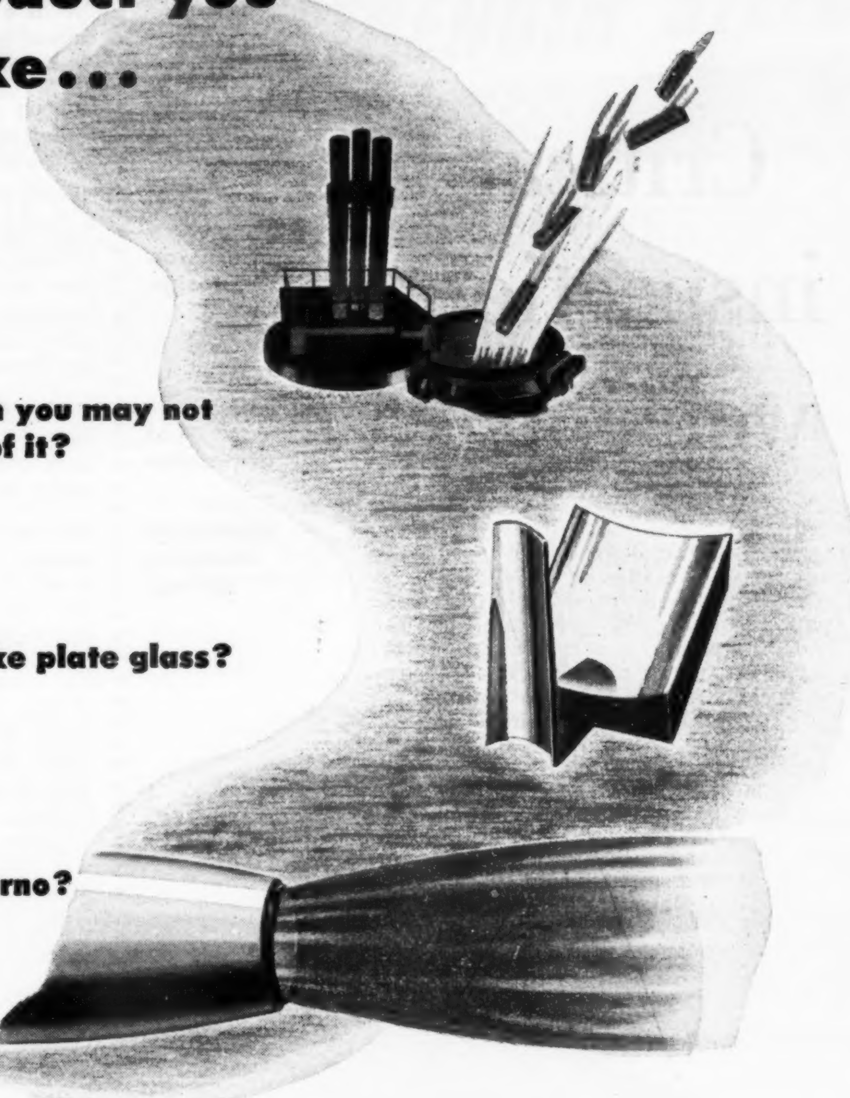
G-E tachometer-generator helps provide accurate speed indication



Whether you are looking for a voltage signal or speed indication on a small machine tool or a large machine operation, G-E tachometer-generators give accurate, dependable responses. In conjunction with suitable indicating instruments, they measure rotating speeds from 1 to 15,000 rpm as well as linear speeds which can be converted to rpm. The precise G-E tachometer-generator (shown above) gives proper voltage signals to the control system.

Each tachometer-generator is self-excited, totally enclosed with ball bearing construction. The over-all compactness of the unit assures fast, simple installations. For full information see GEC-1016A.

GENERAL ELECTRIC



Critical inspection welcomed...

Here's the "workhorse" of the relay industry. Class "A" is a low-cost rugged relay that stands up under critical inspection . . . even after 25 to 50 million operations!

permanently attached contact points

Contact points are resistance-welded—permanently anchored to the spring for the life of the relay. Continual quality checks on the manufacture of the springs assure lifetime service.

exclusive backstop design

Backstop is specially formed to prevent armature freezing and to increase life. A "heavy-duty" armature bearing can be supplied for heavy spring loads or constant high-speed operations; its oversized, hard-metal pin turns in *precision*, long-wearing bearings.

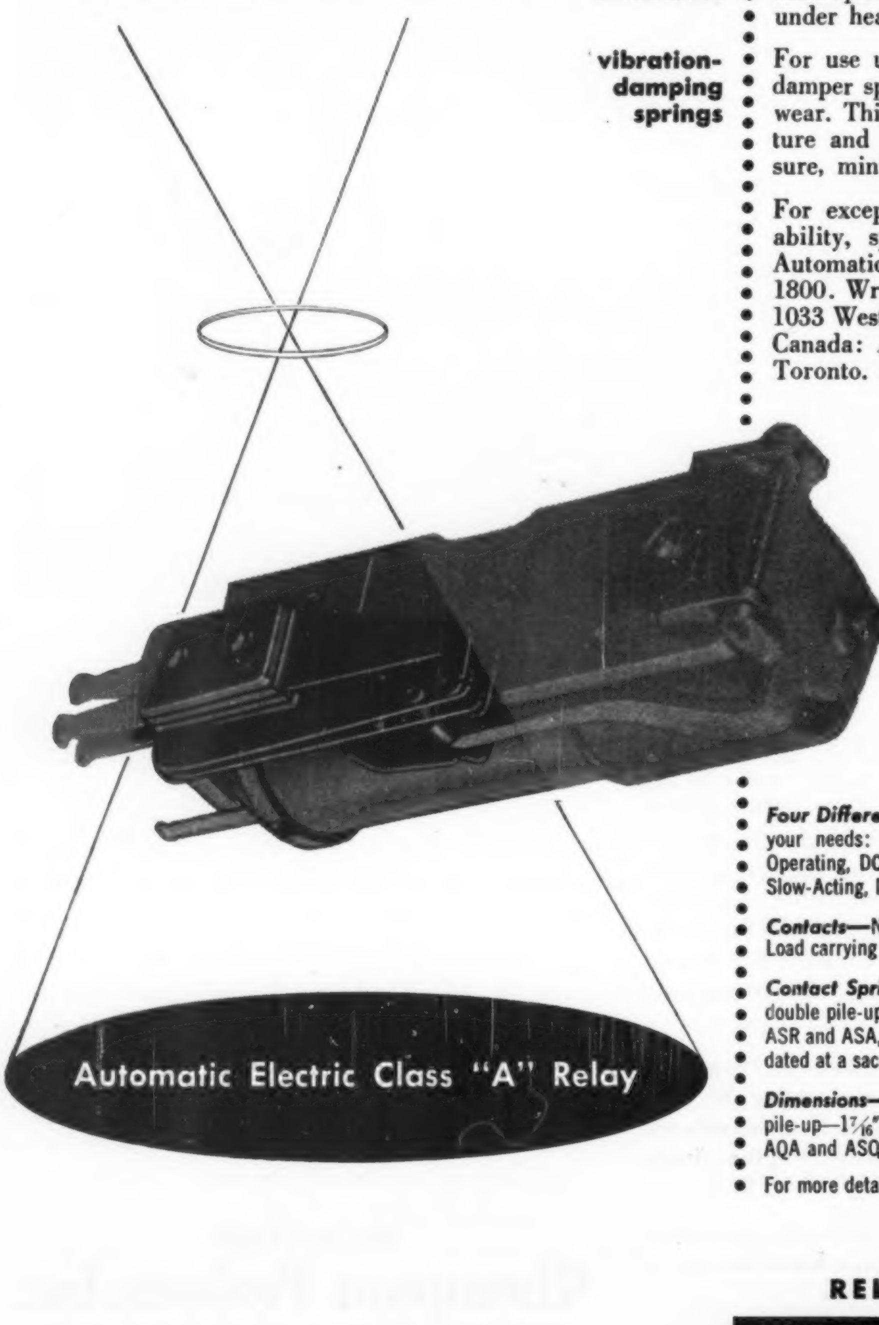
both types of armatures

A choice of "long" or "short" levers gives you the lever ratio required for normal operating and release speeds, or for a residual gap that doesn't vary under heavy-duty conditions.

vibration-damping springs

For use under conditions of extreme vibration, a damper spring can be supplied to prevent excessive wear. This damper spring bears against the armature and the bearing pin with considerable pressure, minimizes movement due to external forces.

For exceptional performance, sensitivity and durability, specify this outstanding member of the Automatic Electric relay family. Ask for Circular 1800. Write: Automatic Electric Sales Corporation, 1033 West Van Buren Street, Chicago 7, Illinois. In Canada: Automatic Electric Sales (Canada) Ltd., Toronto. *Offices in principal cities.*



class "A" specifications

Four Different Types of Class "A" Relays are available to meet your needs: Series AQA—Quick-Acting, DC; Series ASO—Slow-Operating, DC; Series ASR—Slow-Releasing, DC; and Series ASA—Slow-Acting, DC.

Contacts—Normally single, but can be supplied with twin contacts. Load carrying capacity, 150 watts (maximum 3 amps., non-inductive).

Contact Spring Capacity—Can be supplied with single or double pile-ups. Series AQA and ASO, 13 springs per pile-up; Series ASR and ASA, 6 springs per pile-up. (More contacts can be accommodated at a sacrifice of operating speed and release time delay.)

Dimensions—Overall length, 4". Width, single pile-up—1¼", double pile-up—1 7/16". Height (depending upon the number of springs), Series AQA and ASO, 1⅝"-2 3/16"; Series ASR and ASA, 1⅝"-2".

For more detailed information, ask for Circular 1800.

RELAYS SWITCHES

PRODUCTS OF THE INDUSTRIAL DEPARTMENT OF

AUTOMATIC ELECTRIC

CHICAGO

New facts for your file on U·S·S HIGH STRENGTH STEELS

USS HIGH STRENGTH STEELS · UNITED STATES STEEL · TWENTY YEARS OF PERFORMANCE

U·S·S MAN-TEN steel gives heavy-duty earth moving equipment the stamina to stay on the job

● By building maximum strength and durability into parts ordinarily prone to failure through the use of U·S·S MAN-TEN steel, Allis-Chalmers has ensured a high degree of productivity and profitable performance for their TW-300 Motor Wagon.

Used in the sides, ends and bottoms of the bowl of this big capacity, hydraulically operated bottom dump wagon, MAN-TEN steel increases strength 50% over carbon steel construction, provides 50% higher fatigue strength, materially increases resistance to abrasion and to sudden blows in loading. An added advantage is increased resistance to atmospheric corrosion—4 to 6 times that of carbon steel.

As a result, break downs and time-out for repairs are reduced to a minimum, as are maintenance and replacement costs.

In mass-produced truck body members, U·S·S COR-TEN steel saves weight, adds strength, facilitates accurate construction

By mass-producing standardized truck body shapes and sections that can be readily and quickly assembled, Parish Pressed Steel Company, Reading, Pa., has made it possible for local body builders to turn out steel truck bodies to meet individual truck owners' requirements at high speeds and low cost.

The use of U·S·S COR-TEN high strength steel in these prefabricated body sections not only permits light, strong and very durable construction, but because of the high physical properties of this steel and its consistent uniformity, it is possible to hold designs to exact limits and very close tolerances. Over-all results are much better than when less efficient materials are used.

With U·S·S COR-TEN steel construction like this, the body builder benefits because his operations are speeded up and his assembly costs are reduced. The truck owner benefits because he gets a truck that, while light in weight, has maximum strength and durability, is readily repaired if damaged and requires minimum maintenance.

WHAT IS U·S·S COR-TEN HIGH STRENGTH STEEL?

U·S·S COR-TEN is a ductile, low-carbon chromium-nickel-silicon-copper-phosphorous steel having a yield point of 50,000 psi min., and a tensile strength of 70,000 psi min. in thicknesses ½" and under.

Its resistance to abrasion and shock is superior to structural carbon steel; its fatigue resistance—that is, its ability to withstand repeated stresses—is 60% greater.

What particularly distinguishes U·S·S COR-TEN steel is its unusually high resistance to atmospheric corrosion—4 to 6 times that of plain steel, 2 to 3 times that of copper steel. This property helps to assure the long life and low maintenance cost of any equipment, subject to rusting, in which COR-TEN steel is used, whether to obtain greater durability or to reduce weight.

U·S·S COR-TEN steel is produced in all standard products—plates, shapes, bars, sheets, strip, special cold-formed sections and wire. Recommended particularly for application in light and intermediate thicknesses.

UNITED STATES STEEL CORPORATION, PITTSBURGH · AMERICAN STEEL & WIRE DIVISION, CLEVELAND · COLUMBIA-GENEVA STEEL DIVISION, SAN FRANCISCO
NATIONAL TUBE DIVISION, PITTSBURGH · TENNESSEE COAL & IRON DIVISION, FAIRFIELD, ALA. · UNITED STATES STEEL SUPPLY DIVISION, WAREHOUSE DISTRIBUTORS
UNITED STATES STEEL EXPORT COMPANY, NEW YORK

UNITED STATES STEEL

USS

New facts for your file on

U·S·S STAINLESS STEEL

SHEETS · STRIP · PLATES · BARS · BILLETS · PIPE · TUBES · WIRE · SPECIAL SECTIONS

Corrugated sheets of Stainless Steel used for continuous conveyor cover

● Approximately 15,000 pounds of Stainless Steel went into a continuous conveyor cover fabricated by Elwin G. Smith & Company, Pittsburgh, Pa., for Consolidated Edison Company, New York.

The cover consists of 710 corrugated and curved sections each made from a sheet of Type 430 Stainless Steel .029" x 30" x 83". The sheets were corrugated on a standard type machine and then curved to a radius of 2'3" by a roller bending machine.

Officials of the Smith Company are well pleased with the results obtained from their initial use of Stainless Steel in this type of fabrication.

Stainless Steel sheets for continuous conveyor cover are corrugated (below) and then bent to a radius of 2'3" on a roller bending machine (above).

Take advantage of Stainless Steel in your designs and in your selling

Here are more applications typical of the thousands in which Stainless Steel has been used to improve design and add sales appeal. The wide range of valuable properties available in Stainless Steel makes it the ideal material for many, many jobs.

Put Stainless to work for you. It will pay its own way and give you a good return on your investment—especially when it is perfected, service-tested U·S·S Stainless Steel.

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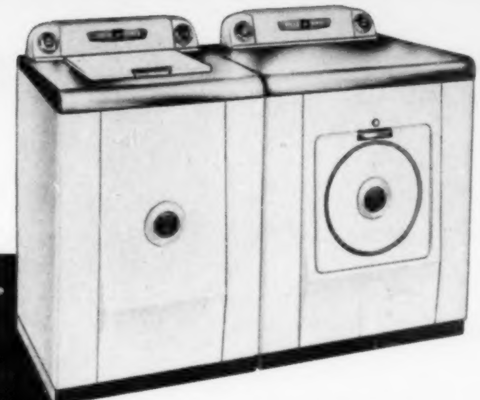
UNITED STATES STEEL

Speed Queen Automatic Washers and Dryers have tops of Stainless Steel



Speed Queen Automatic Washer-Dryer Ensemble.

● The attractiveness, corrosion resistance and long life of Stainless Steel made it the natural choice for the tops of automatic washers and dryers manufactured by Speed Queen Corporation, Ripon, Wis.

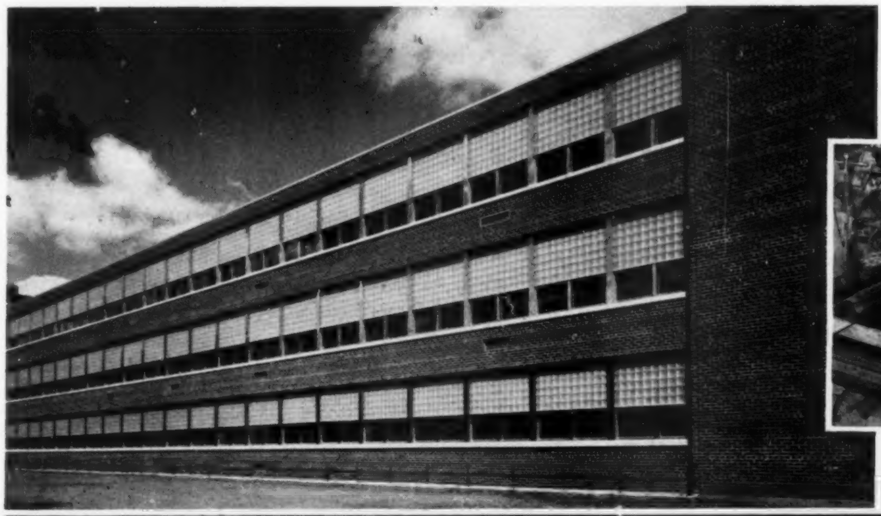
Type 430—a straight-chromium grade of Stainless Steel—is currently being used for this application. Sheets are sheared, punched, drawn, ground and polished into tops that add sales appeal and long life.

Polishing Stainless top for Speed Queen Washer.

Stainless Steel Ribbon Windows highlight this Jamestown, N.D., school

● This grade school building in Jamestown, N. D., is an excellent example of the use of WILKE Stainless Steel Ribbon Windows manufactured by Wilke Metal Products, Inc., Schofield, Wisconsin.

These windows are fabricated from 18-gage Type 430 Stainless. They are brake formed, welded at the corners and then ground and polished to a soft, lustrous finish.



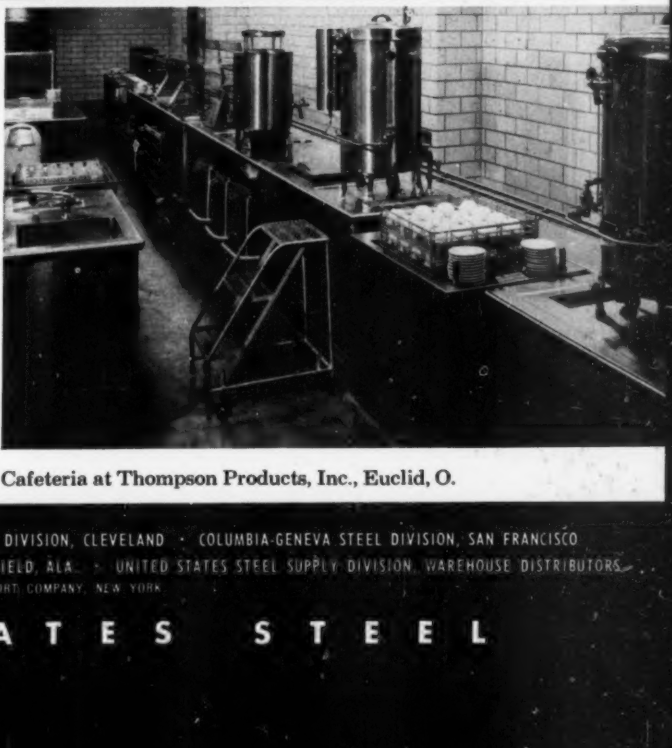
▲ Welding corners of WILKE Stainless Steel Windows.

◀ Ribbon windows in new grade school at Jamestown, N. D.

Award-winning kitchen at Thompson Products literally glistens with Stainless Steel

● Winner of a Merit Award in Institutions Magazine's Seventh Annual Food Service Contest, this new cafeteria at Thompson Products, Inc., Euclid, Ohio, makes full use of the beauty and sanitary qualities of Stainless Steel.

The equipment was "custom-bilt" by Southern Equipment Company, St. Louis, Mo. An interesting feature of this installation is Southern's use of Stainless Steel for the fabrication of the cafeteria counter and various kitchen fixtures.



Cafeteria at Thompson Products, Inc., Euclid, O.

USS

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UNITED STATES STEEL

New facts for your file on

U·S·S CARILLOY STEELS

One source for all your alloy steel plate, sheet, and strip

JUST SPECIFY U·S·S CARILLOY STEELS

All these requirements can be met with flat-rolled U·S·S CARILLOY steels

	PLATE	STRIP	SHEET
METHODS OF MANUFACTURE			
Electric Furnace	x	x	x
Open Hearth	x	x	x
SPECIAL QUALITIES			
Aircraft	x	x	x
Armor	x		
Bearing	x	x	x
Drawing	x	x	x
Flange and Firebox	x		
Metal Cutting Saw		x	x
Razor Blade		x	x
Other Special Qualities	x	x	x
TREATMENTS: INDIVIDUAL OR COMBINED			
Quenched and Tempered	x		
Annealed	x	x	x
Spheroidize Annealed	x	x	x
Normalized	x	x	x
Stress Relieved	x		
CONDITIONS: INDIVIDUAL OR COMBINED			
Hot Rolled	x	x	x
Straight Lengths	x	x	x
Coiled		x	x
Flattened or Leveled	x	x	x
Gas or Special Cutting	x		
Pickled	x	x	x
Sand Blasted	x		
Oiled	x	x	x
Formed, Machined or Other Special Conditions	x	x	x
SPECIFICATIONS REQUIREMENTS INDIVIDUAL OR COMBINED			
Grain Size	x	x	x
Macro-Etch	x		
Micro-Cleanliness	x		
Restricted Chemical Analysis	x	x	x
Special Mechanical Tests	x	x	x
Impacts	x		
Tensile Tests	x	x	x
Bend Tests	x	x	x
AISI-Govt.-ASTM-SAE	x	x	x
Special Specifications	x	x	x

• Note the accompanying list of qualities, treatments, conditions and specification requirements that can be furnished in U·S·S Carilloy plate, sheet and strip. This is the widest selection of flat-rolled Alloy Steel products you can secure from any one producer. Our unmatched mill flexibility and size range enable you to order anything from a razor blade strip to a plate for a battleship. This streamlines your purchasing, assures consistent quality and simplifies your manufacturing problems.

For complete information, contact our nearest sales office, or write direct to United States Steel Corporation, Room 2819-Z, 525 William Penn Place, Pittsburgh 30, Pa.

Free metallurgical assistance on any steel problem

Any time you have a metallurgical or fabricating problem, call in a United States Steel Service Metallurgist. He has an extensive knowledge of all types of Alloy Steels and can help cut costs by offering suggestions to assist your engineering and production people.

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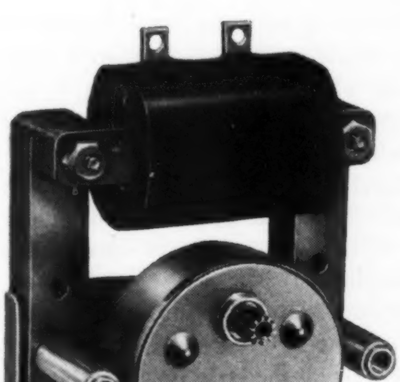
UNITED STATES STEEL

FILLING STATION

...FOR LONG-LIFE LUBRICATION

Telechron Synchronous Timing Motors



MODEL H-10. *Low-cost, light-duty motor capable of handling high momentary peak loads. Ideal for washing machines, dishwashers, refrigerators, and other appliance timer uses.*

Seal the oil in with the moving parts, and they'll both last longer. That's the simple reason behind the long, dependable life of Telechron Synchronous Timing Motors.

Into each motor goes a measured amount of special oil—carefully formulated for the particular service the motor is to perform. Then the unit is sealed. Dirt and dust can't get in. Lubricant is lifted by capillary action from the reservoir to all bearings, and flows continuously to all gears, efficiently . . . so efficiently, in fact, that many Telechron motors are still operating accurately and dependably after 20 years of continuous use.

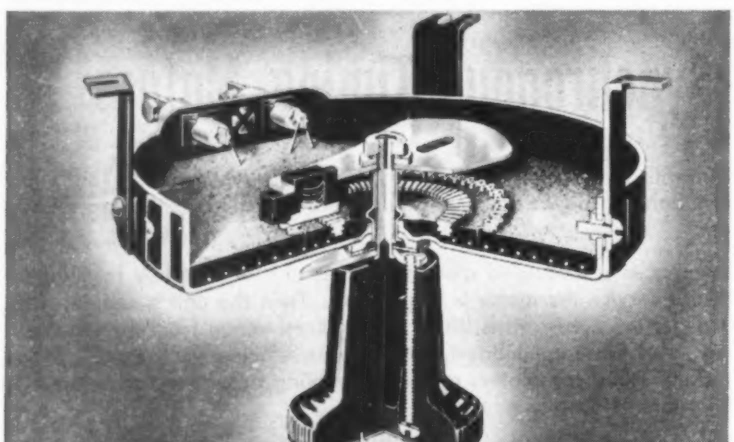
There are other advantages, too, in Telechron Synchronous Timing Motors. Quick starting, due to the lightweight rotor. Power-line accuracy, because of true synchronous operation. Cool running, with the field coil isolated from the rotor unit. Altogether, a combination of worth-while features unique in the field of electric timing.

Telechron motors are available in a wide range of speeds and torque ratings, and for any standard AC power source. Get full details. Write Telechron Department, General Electric Co., 212 Homer Ave., Ashland, Mass.

Telechron®

MARK OF TIMING LEADERSHIP

EASTMAN KODAK'S RECORDAK MICRO-FILE MACHINE can microfilm 32 acres of valuable records for storage in only 12 cubic feet of filing space. Accurate performance of Ward Leonard Vitrohm rheostat (in circle) helps to assure clear, sharp, reproduction on microfilm.



CUTAWAY VIEW OF TYPICAL WARD LEONARD RHEOSTAT. Eastman Kodak's Recordak engineers chose a 13-in. Vitrohm rheostat to control lamp voltage in the Recordak Micro-File machine. It provides 161 graduated steps of smooth, accurate, dependable control.

Recordak Corporation uses Ward Leonard rheostat for accurate light control in microfilming

• Light plays a critical role in committing billions of industry's valuable records to microfilm every year. Its intensity must be carefully controlled for each exposure on Eastman Kodak's Recordak Micro-File machine in order to make proper allowances for paper texture, color and reflectivity.

To handle the job of adjusting lamp voltage to meet these varying requirements, Recordak engineers selected a dependable 13-inch Ward Leonard rheostat. It's the only adjustable control used in operating the unit.

Accurate performance of this rheostat in Recordak Micro-File units helps to assure clear reproduction of every detail in the documents being photographed.

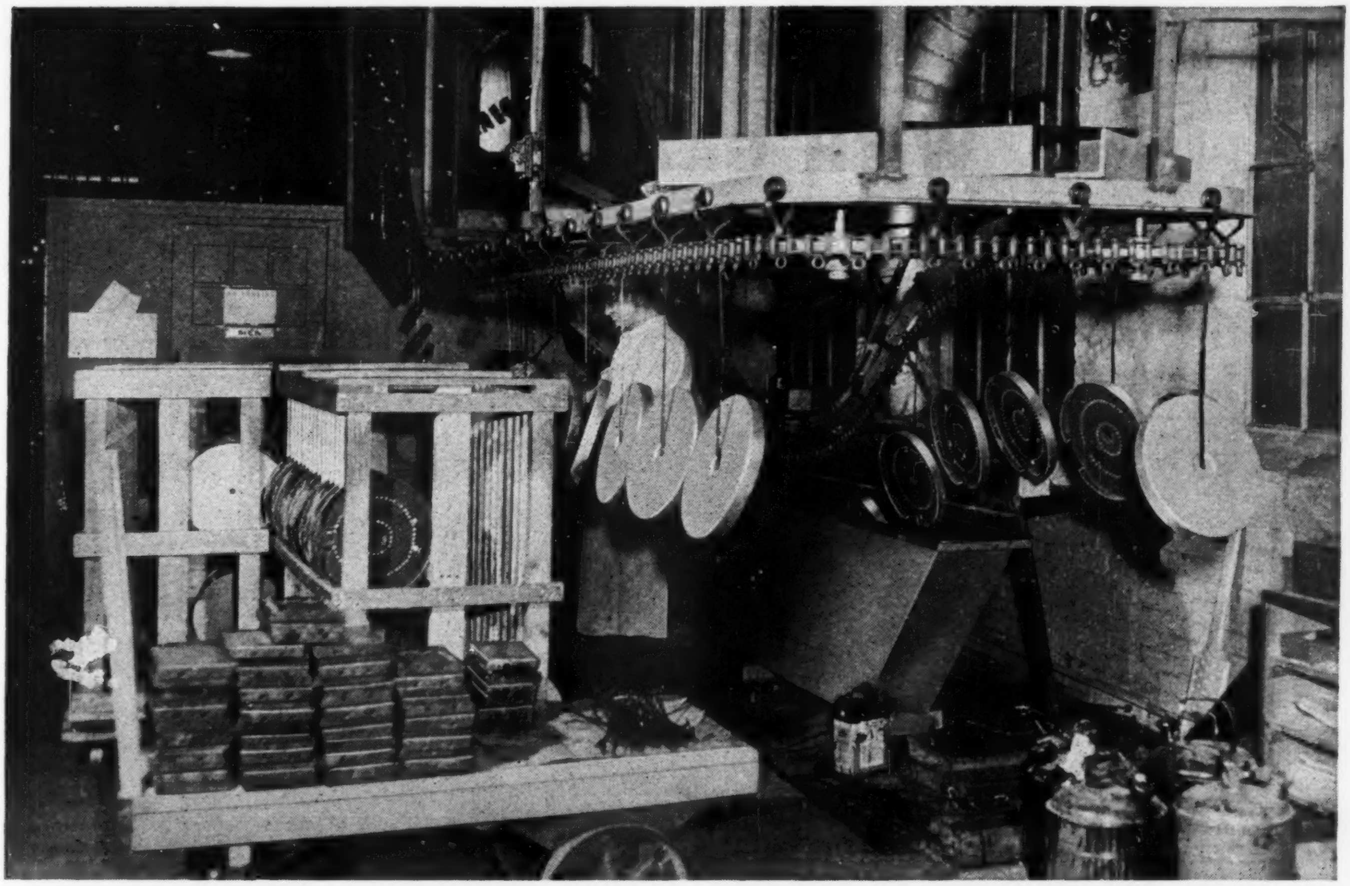
If your product requires accurate electrical controls, our engineers are ready, willing and able to help you select the right ones for the job. Write Ward Leonard Electric Co., 58 South St., Mount Vernon, New York.

WARD LEONARD

WARD LEONARD ELECTRIC COMPANY

MOUNT VERNON, NEW YORK

Result-Engineered Controls Since 1892



HEAT-RESISTANT FINISH is automatically applied and infra-red baked on all Vitrohm pressed steel rheostat plates.

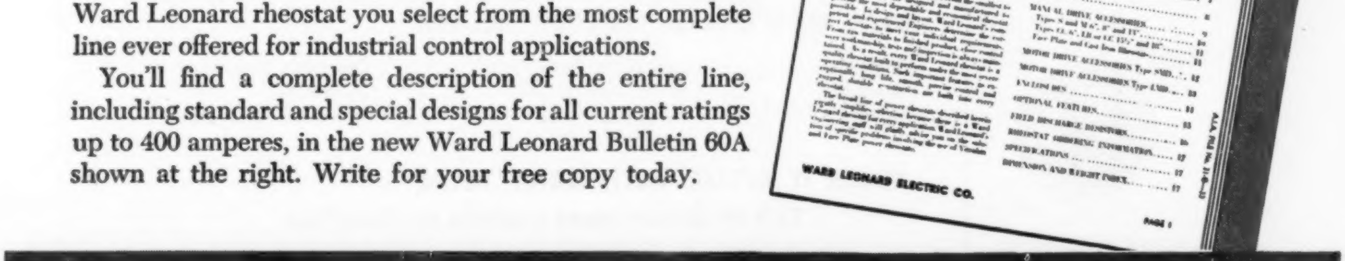
Here's how dependable performance is built into every Ward Leonard rheostat

• All rheostat materials, from vitreous enamel frit to heat-resistant finish, as well as all manufacturing processes, are carefully controlled by Ward Leonard engineers.

After assembly, rheostats are given thorough mechanical and electrical tests to guard against any constructional defects that might impair operating life or accuracy.

That's why you can depend on the performance of any Ward Leonard rheostat you select from the most complete line ever offered for industrial control applications.

You'll find a complete description of the entire line, including standard and special designs for all current ratings up to 400 amperes, in the new Ward Leonard Bulletin 60A shown at the right. Write for your free copy today.



Production costs cut 90%

WITH GRAY IRON

Gear cover fabricated from nine pieces.
Arrows show cut-outs.

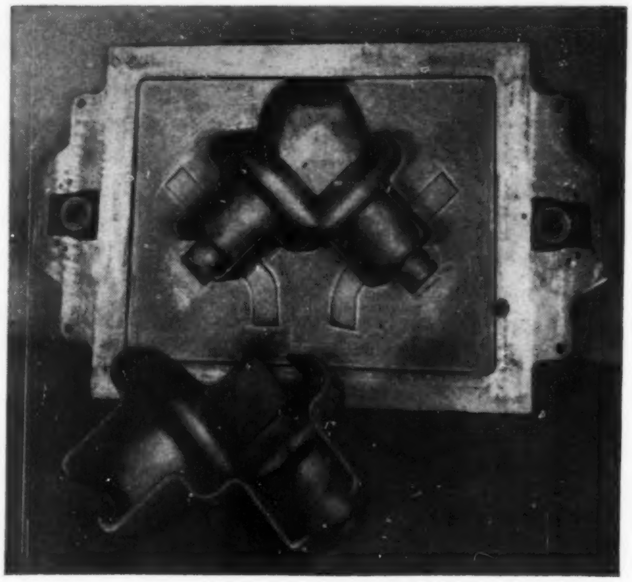
Made entirely in green sand, the Gray Iron castings were produced at one-tenth the cost of fabricated gear cover.

Formerly, nine separate pieces were shaped and welded into a fabricated gear cover for a braiding machine. Then, the manufacturer redesigned the gear cover for production in Gray Iron.

The finished and machined casting cost *less than 10%* of the production cost for the fabricated piece. This saving quickly paid for pattern-equipment costs . . . netting a *90% saving* on all parts thereafter. By redesigning for Gray Iron, the manufacturer not only made substantial savings, but obtained the advantages of a streamlined, vibration-absorbing gear cover.

Look at the structural and functional requirements of your fabricated parts. You can undoubtedly use the unique characteristics of Gray Iron—and the economy offered by this casting process. Why not write us for more information?

Matchplate and gear cover.



GRAY IRON CHARACTERISTICS INCLUDE:

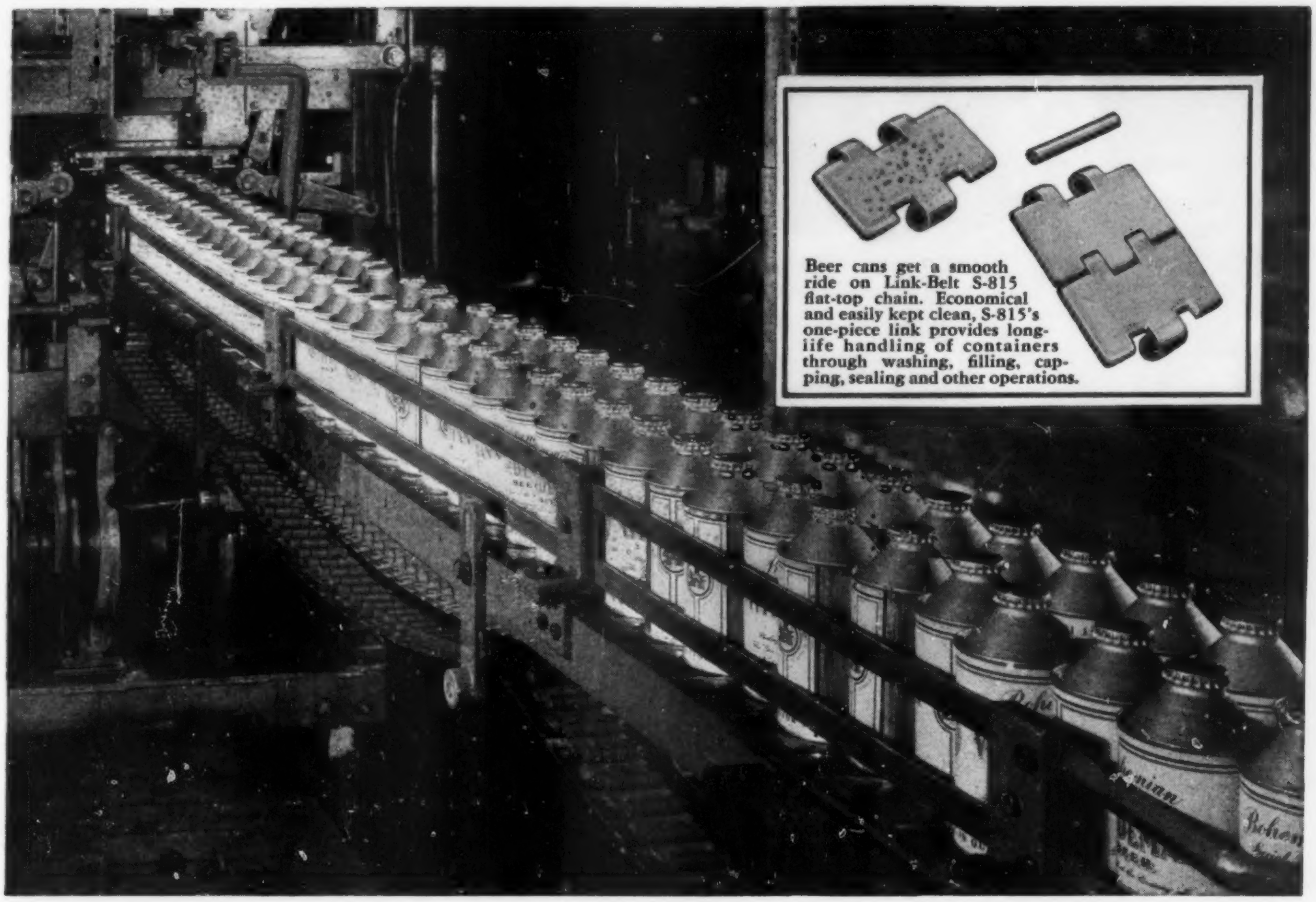
- Castability
- Strength
- Rigidity
- Low Notch Sensitivity
- Wear Resistance
- Heat Resistance
- Machinability
- Corrosion Resistance
- Durability
- Vibration Absorption

MAKE IT *BETTER* WITH GRAY IRON

75% of all cast metal products are Gray Iron

GRAY IRON FOUNDERS' SOCIETY, INC.

NATIONAL CITY-E. 6th BLDG., CLEVELAND 14, OHIO



Here's a smooth-operating, flat-top conveyor chain...moderately priced

LINK-BELT offers the chain that's best for every job

THE many varied drive and conveying jobs performed by chains and sprockets require different physical characteristics. The S-815 flat-top conveyor chain is an example of how Link-Belt builds a type and size for every purpose.

In addition, every chain in the complete Link-Belt line is built for longer life. Rigid control of raw materials and manufacturing processes is your assurance of uniformity.

For all the facts on S-815 flat-top conveyor chain, ask for Folder 2344. And for information on the complete Link-Belt chain line, see your nearest Link-Belt district sales representative.

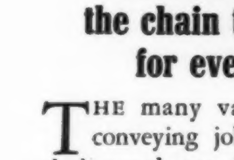
LINK-BELT COMPANY: Plants: Chicago, Indianapolis, Philadelphia, Colmar, Pa., Atlanta, Houston, Minneapolis, San Francisco, Los Angeles, Seattle; Scarboro, Toronto and Elmira, Ont. (Canada); Springs (South Africa); Sydney (Australia). Sales Offices in Principal Cities.

LINK-BELT

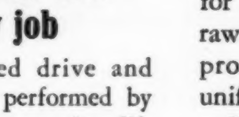
CHAINS AND SPROCKETS

13,325

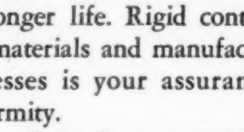
No one chain serves every purpose . . . get the RIGHT one from Link-Belt's complete line



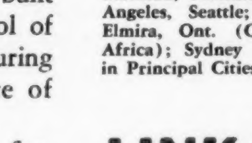
Class RC crescent flat-top chain—travels in horizontal planes, around corners; ideal for bottles and cans.



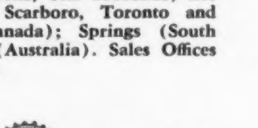
Malleable roller chain for conveyors or inclined elevators where reduced chain pull is desired.



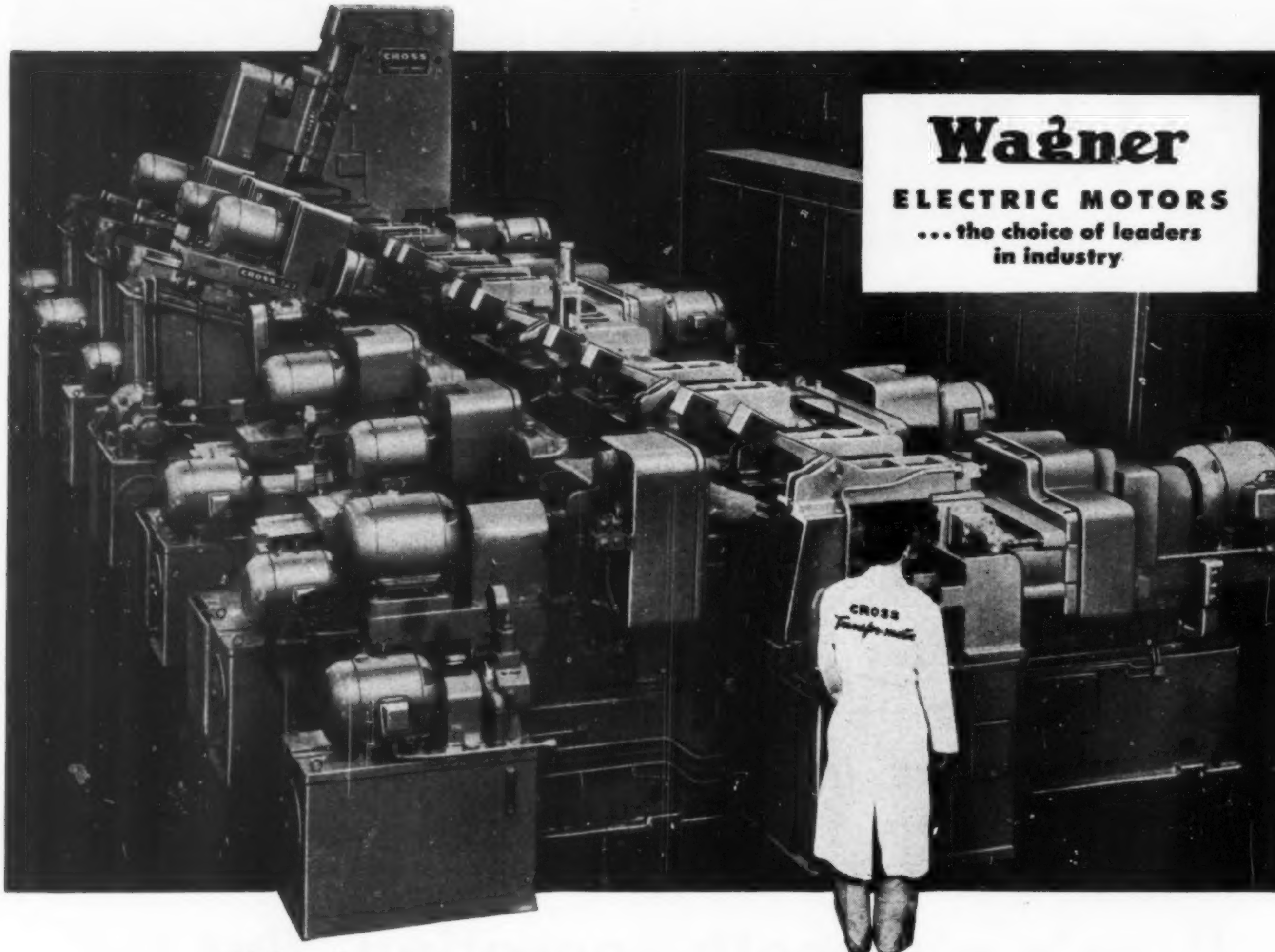
Class 400 swivel chain—carries cartons, cases, etc. over paths with both vertical and horizontal turns.



Class RC flat-top chain—precision made for smoothest, spill-free carrying and long life.



Class RC universal crescent flat-top chain can be used for multiplane operation traveling variety of paths.



Wagner Motors provide the power to machine 72 tractor cylinder blocks per hour

This Cross Transfer-matic was built for Ford Motor Company for use in its Tractor and Engine Division. It mills, drills, bores, taps, reams and chamfers 72 tractor cylinder blocks in an hour, and is equipped throughout with Wagner totally-enclosed motors. (A total of 32 motors, ranging from ¾ to 15 hp.)

Machine tools powered with Wagner totally-enclosed motors are assured freedom from excessive downtime caused by motor failure, because these motors are fully protected against damage from steel-filings, chips, dust, dirt, fumes and moisture.

For your requirements, there is a Wagner Motor to fit every need—a complete line for all current specifications, with a wide variety of enclosure types and mountings. Bulletin MU-185 gives full information. Do you have a copy?

A nearby Wagner engineer will be glad to help you select the *right* motors for your next application. Consult the nearest of our 32 branch offices, or write us.

Wagner Electric Corporation EST. 1891

WAGNER ELECTRIC CORPORATION
6404 Plymouth Ave., St. Louis 14, Mo., U.S.A.

ELECTRIC MOTORS • TRANSFORMERS • INDUSTRIAL BRAKES
AUTOMOTIVE BRAKE SYSTEMS — AIR AND HYDRAULIC

BRANCHES IN 32 PRINCIPAL CITIES

M53-17


U. S. Air Force crash trucks and rescue vehicles by American-LaFrance-Foamite Corp., Elmira, N. Y., use Revere Aluminum.
Aluminum Conductor Busway, light and economical, by Bull Dog Electric Products Co., Detroit, Mich., uses Revere Aluminum (EC Grade) Bar.
Industry Depends Upon The Versatility Of
REVERE ALUMINUM
Revere meets the requirements of industry with a wide range of aluminum alloys in extruded shapes, coiled sheet, drawn tube, electrical bar and forgings. And industry knows the fine, uniform quality of Revere Aluminum. Each customer's requirements, based on their specific uses of Revere products, receives careful attention by Revere quality control.
For assistance in the efficient manufacture of products using aluminum, you are invited to utilize Revere's Technical Advisory Service.
REVERE ALUMINUM
REVERE COPPER AND BRASS INCORPORATED
Founded by Paul Revere in 1801
230 Park Avenue, New York 17, N.Y.
Mills: Baltimore, Md.; Chicago and Clinton, Ill.; Detroit, Mich.; Los Angeles and Riverside, Calif.; New Bedford, Mass.; Rome, N. Y.—Sales Offices in Principal Cities.
SEE "MEET THE PRESS" ON NBC TELEVISION, SUNDAYS
Many makers of indoor and outdoor chairs and furniture use Revere Aluminum Tube. This chair by Lawnlite Co., Miami, Florida.
The heart of this cylinder surfacing hone is a Revere Aluminum Extruded Shape. Photo courtesy Ammco Tools, Inc., North Chicago, Ill.
A Revere Aluminum Extruded Shape is used by the meat packing industry as smoke-sticks for hanging meats during the smoking process. Photo courtesy The Globe Co., Chicago, Ill.

NATIONAL OIL SEAL LOGBOOK

Write our Redwood City office for reprints of this Logbook page

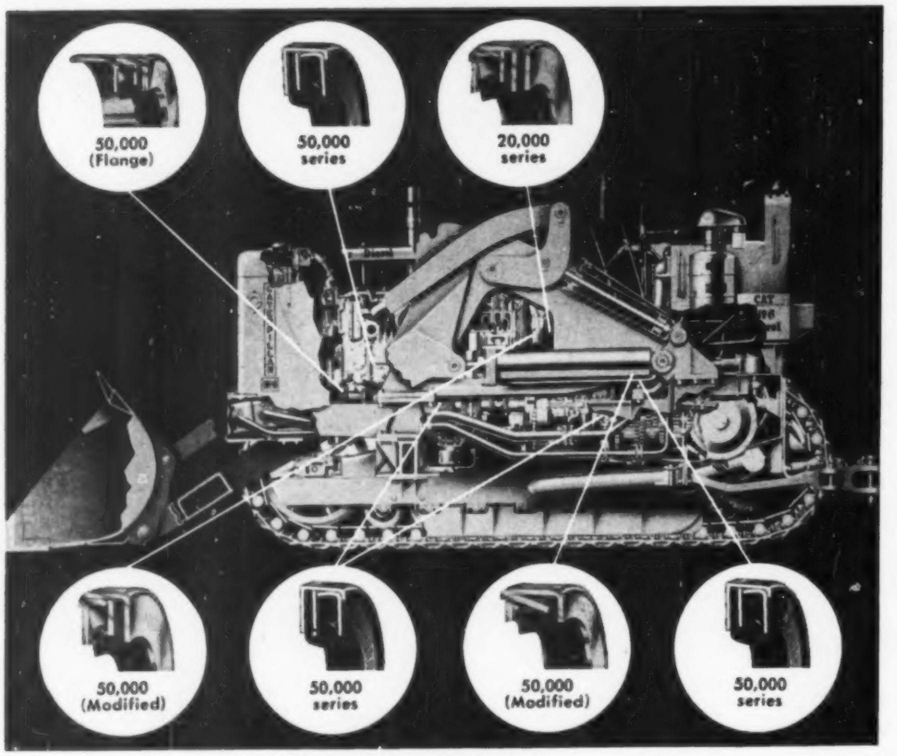


Figure 1. Caterpillar No. 6 Shovel

National Oil Seals used at key points throughout new Caterpillar No. 6 Shovel

The Caterpillar No. 6 Shovel is a 66 hp heavy-duty unit of two-yard capacity, with a non-oscillating track frame. As in other Caterpillar equipment, National Oil Seals are installed at key points to retain lubricant, exclude dirt and water, and prolong life of bearings and assemblies.

Among the many positions where National seals are installed is the front crankshaft bearing. Here a National 50,000 series seal with a special, press-fit mounting sleeve is employed. This seal is of single-lip design, and has an accurately spring-tensioned leather sealing member to retain crankcase oil.

On the fuel pump shaft, Caterpillar also uses a National 50,000 seal with a spring-tensioned leather sealing lip. On the starting engine crankshaft (rear) a National 20,000 series seal is used. This is a dual-lip seal, employing a leather main sealing lip to retain lubricant and a springless leather auxiliary lip as a dust excluder.

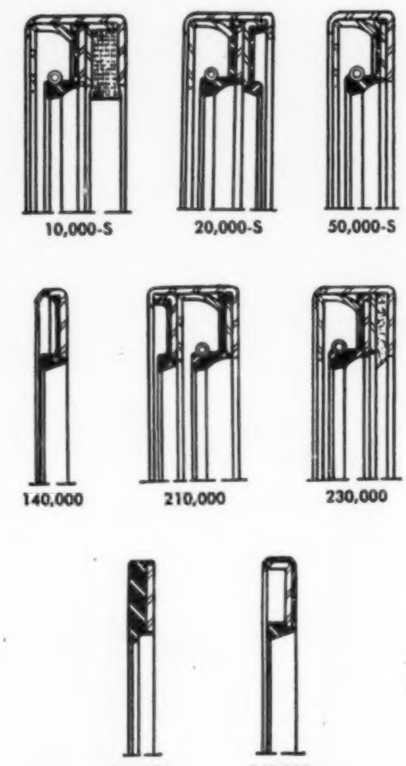
At other vital positions, including the starting engine crankshaft (front), track carrier rollers, the upper transmission shaft, gear shifter shaft and shaft housing, modified or standard-design National 50,000 series leather seals are employed. National precision shims are also utilized extensively in the shovel, and National O-Rings are used in the steering clutch control and dump control assemblies.

National seals used in the Caterpillar No. 6 Shovel are all basic designs or modifications. National offers over 2,500 standard-design seals; can also provide special seals for special requirements. Your National Field Applications Engineer has complete information.

*T. M. Reg.

Sealing News & Tips

National Syntech* Oil Seals are widely used for operation in applications where intermittent temperatures reach 250° F, speeds reach 3,600 F.P.M., and total indicator runout reaches .030. Basic designs shown here employ tough, accurately manufactured steel outer cases, precision-tensioned springs and accurately molded and trimmed Syntech sealing lips unaffected by water, most lubricants, detergents or industrial fluids.



"Let Your Decision be Based on Precision"

NATIONAL OIL & GREASE SEALS
O-RINGS SHIMS

NATIONAL MOTOR BEARING CO., INC.
General Offices: Redwood City, California
Plants: Redwood City, Calif.; Downey (Los Angeles County), Calif.; Van Wert, Ohio

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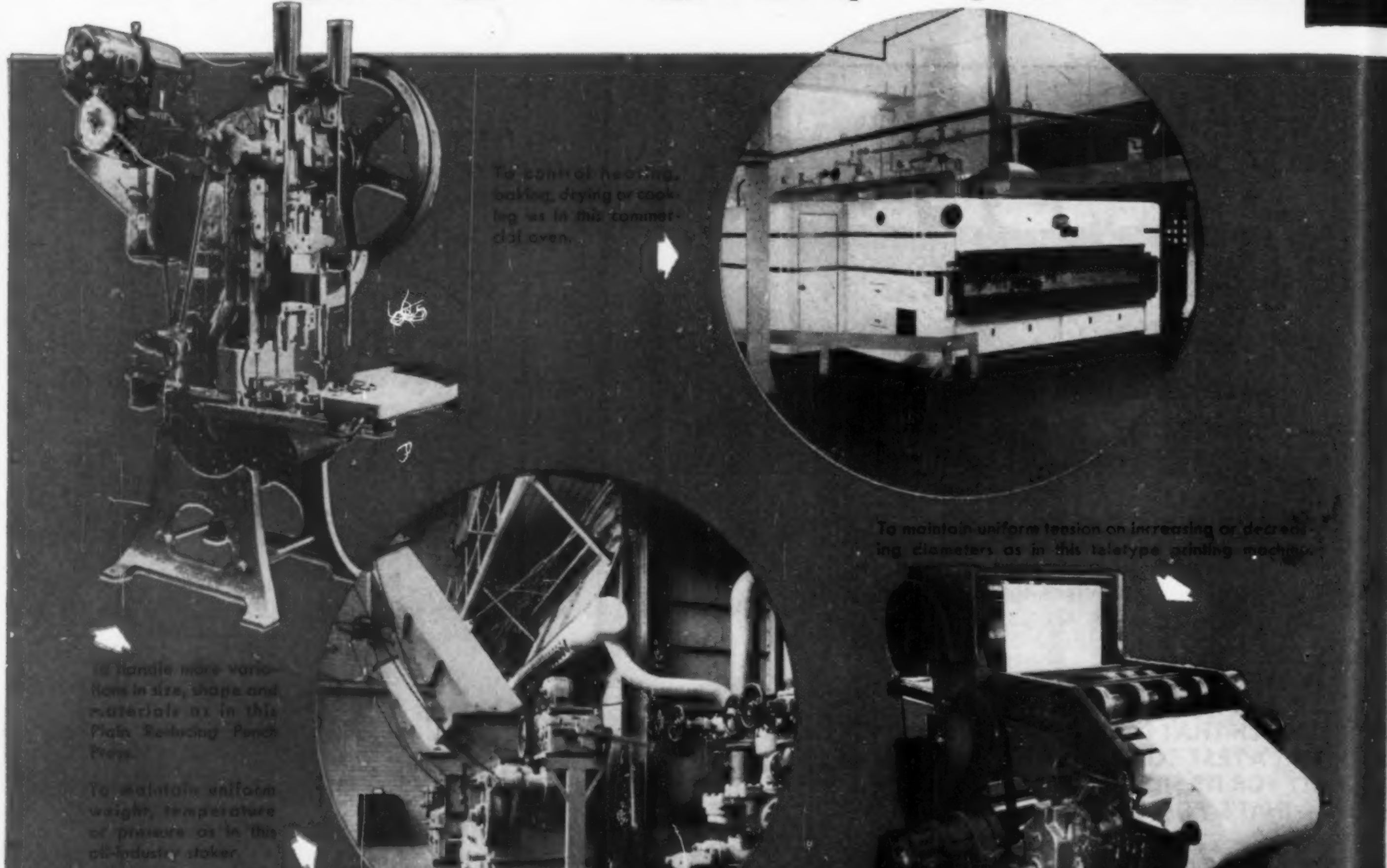

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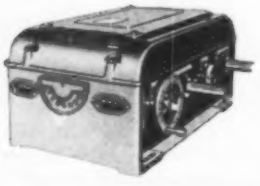

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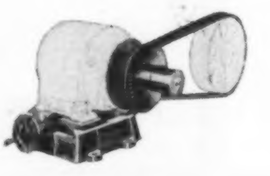
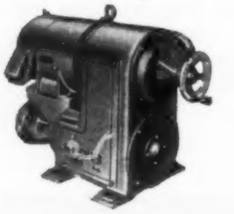


Vari-Speed Motor Pulley converts any constant speed motor to an accurate, variable speed drive with 4:1 ratio. Rugged, dependable—sizes to 10 hp. VARI-SPEED JR. available for light machinery.



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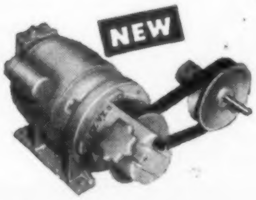

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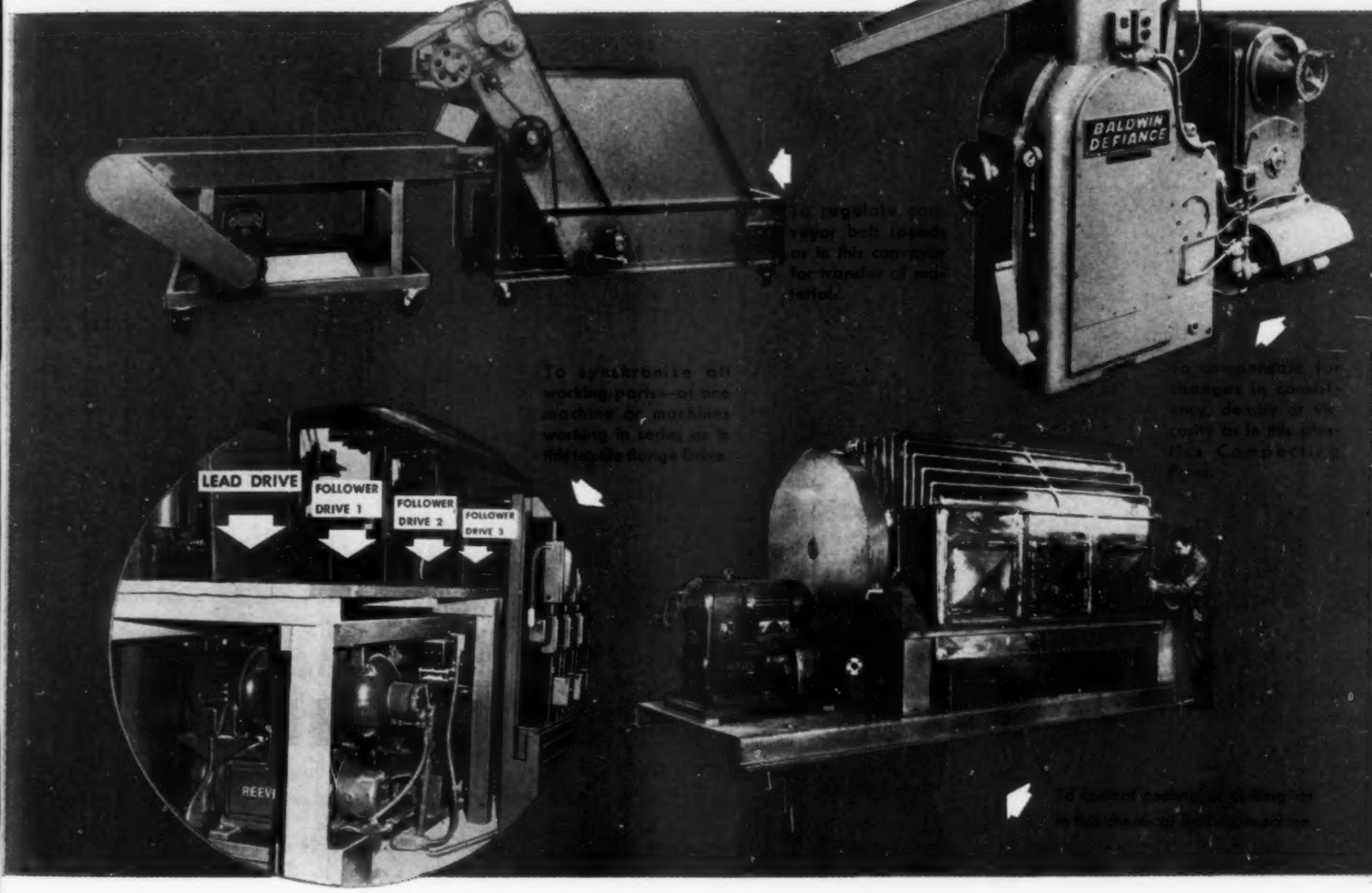

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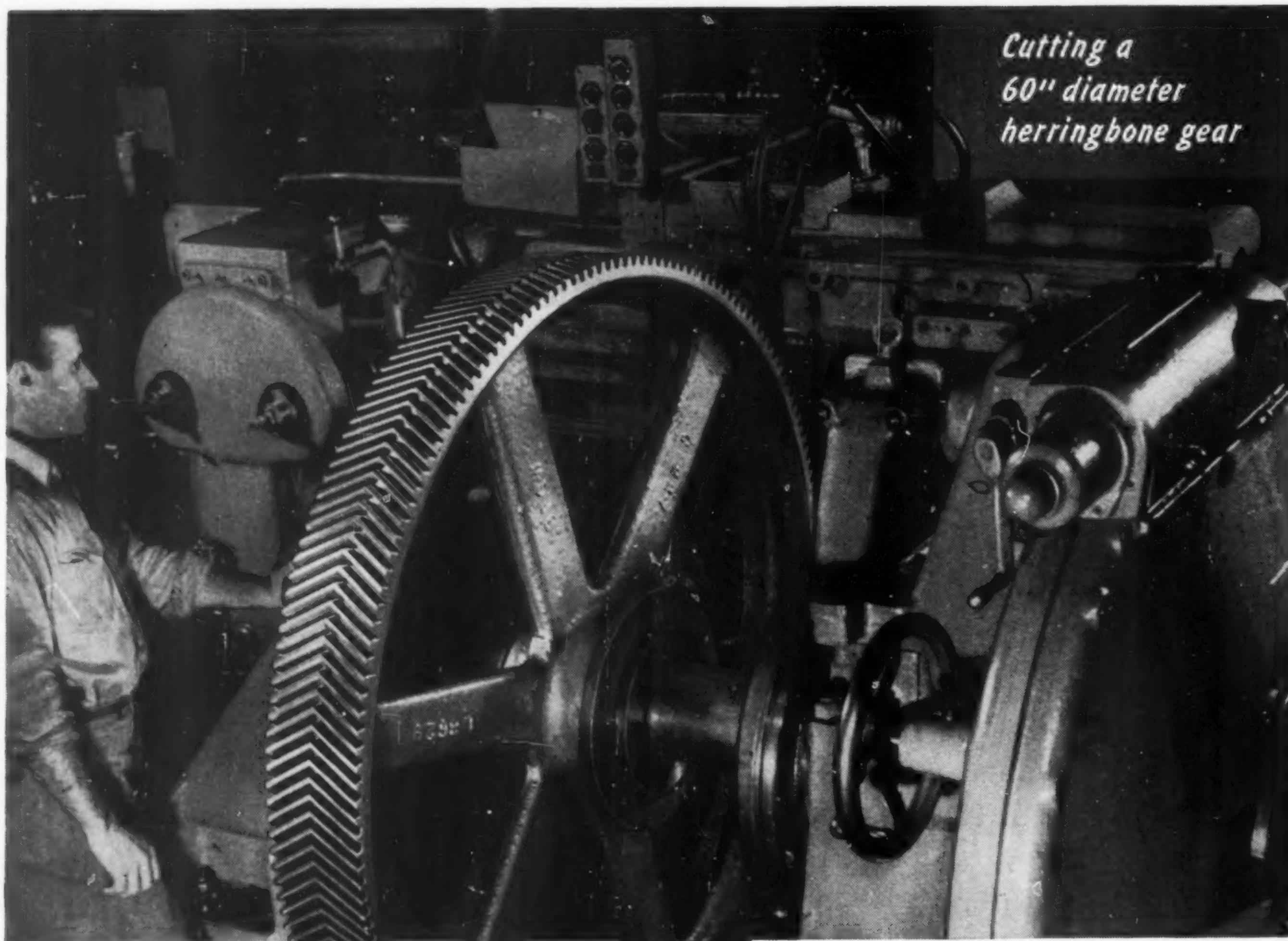

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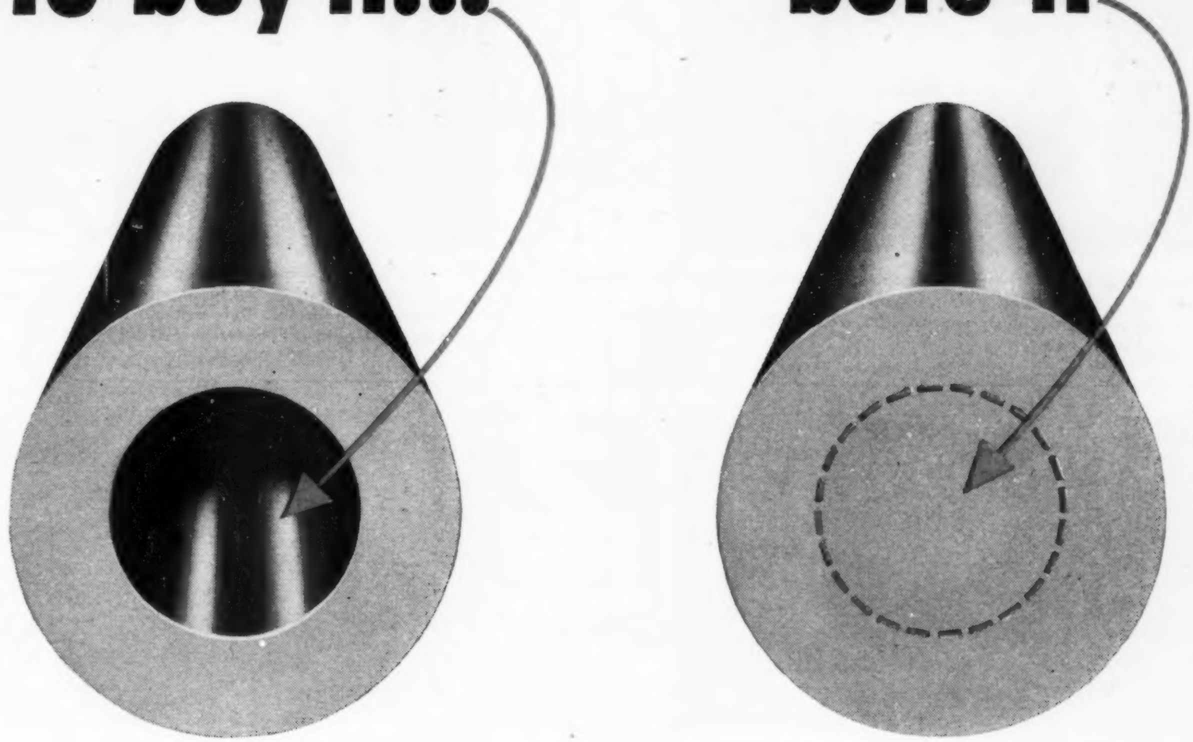

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To help you save even *more* steel, the Timken Company's engineers will study your problem and recommend the most economical tube size for your hollow parts job — guaranteed to clean up to your dimensions.

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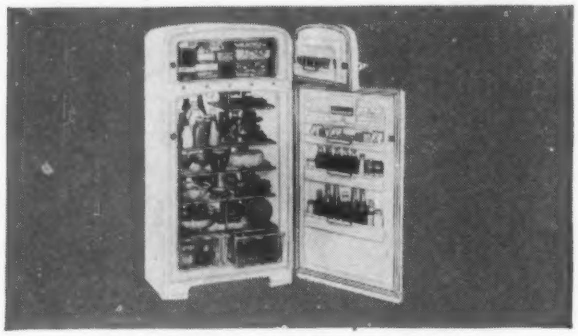

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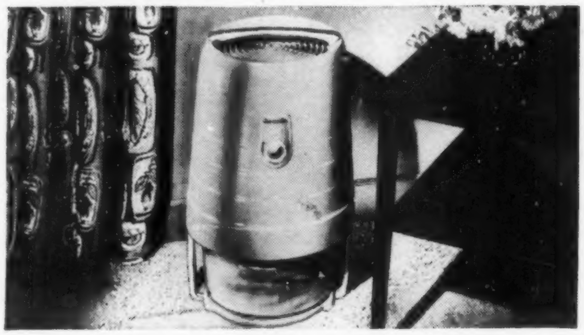
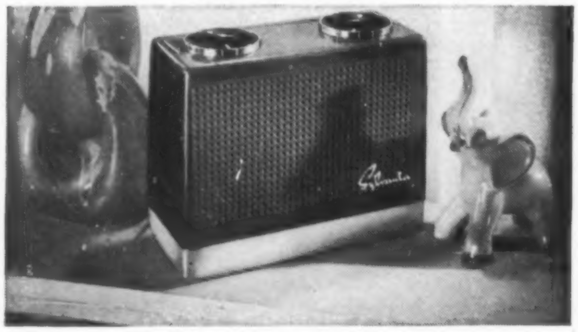


You needed a plastic with high impact, high heat resistance, fast setting, and an excellent finish for moisture conditioner housings. Your demand was answered with Koppers Modified Polystyrene MC-309.



You needed a medium impact, high heat-resistant plastic with high lustre . . . for radio cabinets where breakage and scuffing on ejection presented a reject problem. Koppers Modified Polystyrene MC-409 had the exact characteristics to do the job.

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MC-309	High Impact, Highest Heat Distortion Temp.
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MC-409	Medium Impact, Highest Heat Distortion Temp.

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DESIGNING WITH ALUMINUM

This is one of a series of information sheets which discuss the properties of aluminum and its alloys with relation to design. Extra or missing copies of the series will be supplied on request. Address: Advertising Department, Kaiser Aluminum & Chemical Sales, Inc., 1924 Broadway, Oakland 12, California.

NO. 5

STRENGTH AFTER WORKING

RATES OF GAIN IN YIELD AND TENSILE STRENGTHS VARY FOR DIFFERENT ALLOYS

WROUGHT ALUMINUM alloys are being increasingly used for formed pieces, parts and articles where specified strength is a prime requirement. This makes more important the development of quantitative data concerning the increase in yield and tensile strengths that occurs during forming operations.

The phenomenon of strain or work hardening, of course, has long been known. It applies to both the non-heat-treatable and heat-treatable aluminum alloys. But its influence is of major practical concern only in the case of the non-heat-treatable (sometimes loosely called "common") alloys because the final strength of a product formed from them is determined by that of the original material plus whatever is added by the working of the metal during fabrication.

The non-heat-treatable alloys are logically the first to be considered, and are in fact most generally used in applications where forming requirements are moderate to severe. Where development of their fullest strength is desired, it is frequently possible to start with metal in an intermediate, rather than the annealed, temper if the required forming is not too severe.

Research Tests Show Changes from Stretching

Tests conducted by Kaiser Aluminum's Division of Metallurgical Research have produced interesting and significant data concerning the relative in-increases, after varying amounts of stretching, in both yield and tensile strengths and the rate of work hardening for four of the alloys usually used for forming. These data may be helpful as a guide in both design and alloy selection.

Specimen strips of .040" thick 2S, 3S, 50S and 52S sheet were first tested in the annealed ("O" temper) condition, then stretched specified amounts in a Baldwin-Southwark testing machine and tested again after stretching. The tests were made with the grain of the metal, and in the same direction in which the samples were stretched.

Fig. 1. Effects of stretching upon yield strength.

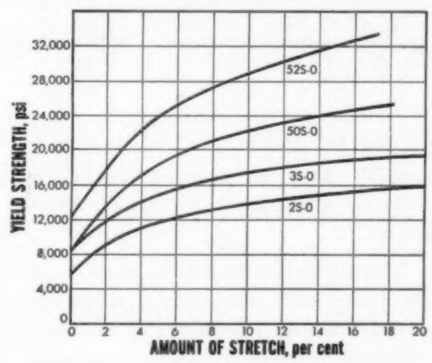


Fig. 2. Effects of stretching upon ultimate strength.

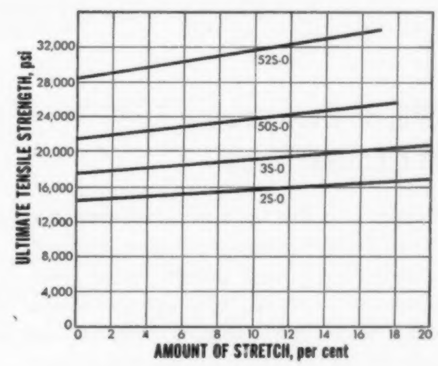


Fig. 3. Rate of work hardening during stretching. Curves plotted from tangents to curves in Fig. 1.

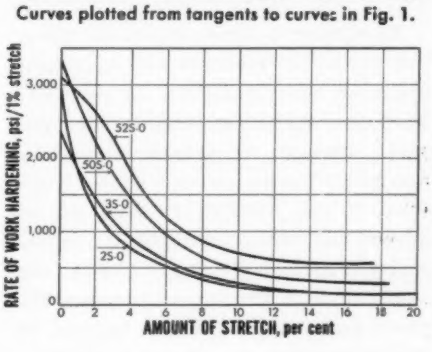
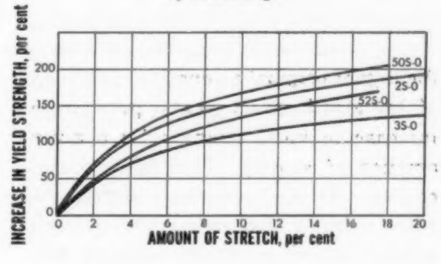


Fig. 4. Increases in yield strength caused by stretching.



The results are shown in Figs. 1 to 4. These curves are plotted in terms of yield and tensile strength after stretching, the relative percentage increase in yield strength for a given amount of stretching and the rate of work hardening.

Emphasis was placed on the changes in yield strength because of its importance as a guide to dent resistance in many formed products, where denting frequently is a cause for rejection.

The relationships between these four alloys as shown in the charts are of particular interest when they are considered in conjunction with cost, fabrication problems and service requirements. For instance, it is clear why use of 2S, although it is the most economical of these alloys and possesses high ductility, is declining except where there are minimum strength requirements. It is evident as well that good strength in the finished product may be obtained with 52S, although its more rapid work hardening rate indicates the fabrication considerations which may be involved by comparison with 3S and 50S, whose work hardening rates and strengths are more moderate.

PLEASE TURN TO NEXT PAGE ➡

TABLE 1 Comparison of Properties in Crisper Pans Drawn from .040" 3S-0 and 50S-0 Sheet						
Alloy	Location	Thickness, inches	Direction of Test	Ultimate Strength, psi	Yield Strength, psi	% Gain in Yield Strength For 50S
3S	Flanges	.0426	------------	18500	17700	
50S	Flanges	.0440	------------	25600	24500	38
3S	Ends	.0373	With grain	17250	16400	
50S	Ends	.0386	With grain	25700	22300	36
3S	Ends	.0346	Across grain	17950	17700	
50S	Ends	.0352	Across grain	27080	25100	42
3S	Sides	.0338	With grain	20000	19200	
50S	Sides	.0349	With grain	27550	26100	36
3S	Sides	.0375	Across grain	17625	16750	
50S	Sides	.0385	Across grain	25750	23200	38

50S, 3S Performance in Identical Piece

Along with the tests of specimens described above, tests were also made of production-formed pieces made from alloys 50S and 3S. The test specimens were taken both with and across the grain from identical refrigerator crisper drawers drawn from .040" sheet in each case.

The results, tabulated in Table 1, demonstrate the advantage obtained in this application deriving not only from the higher original strength of 50S over 3S but also from its greater percentage increase in yield strength for a given amount of stretch. This latter factor can be of decided advantage in strengthening sections of an item which by their nature receive minimum working during fabrication.

Depending upon the location from which the specimens were taken, the yield strength due to forming was from 36% to 42% greater for the 50S drawer, a substantial gain over the increase in yield strength of the 3S pan.

These tests all serve to re-emphasize the importance of giving careful consideration to all factors involved in selecting the aluminum alloy for a particular application, as well as the versatility provided by the entire family of aluminum alloys.

Many Alloys Provide Good Formability

Aluminum is generally known as one of the most easily formed metals. An accurate measure of formability has yet to be developed, although it is possible to draw general conclusions from an analysis of mechanical properties.

The spread between the tensile and yield strengths, and the per cent elongation provide indications of the formability of a particular alloy. In general, the more severe the operation, the greater both these measures should be.

Elongation is a measure of ductility, but cannot be considered by itself as a criterion of formability in practice. Material must be stressed beyond the yield strength to deform plastically and retain the desired new shape. However, if the forming stresses should exceed the tensile strength, the material will rupture (fracture): hence, the importance of the spread between the yield and tensile strength values. In this connection it is often found that a stronger temper of the same alloy may provide better formability because of its ability to withstand higher forming pressures.

Most of the generally used wrought aluminum alloys possess good spreads and elongation in their annealed (O) tempers. This is true of the heat-treatable alloys as well as the non-heat-treatable alloys.

It is not to be overlooked, however, that even complicated forming can be and is being done regularly with the heat-treatable aluminum alloys, including those of the highest strength, after solution heat treatment but before aging to maximum strength. This is possible because after the quenching that concludes solution heat treatment the heat-treatable alloys remain quite soft for varying periods of time. Within a few minutes the alloying constituents begin to precipitate as extremely small particles in the solid metal, and the strength at room temperature begins to increase. Some alloys, 24S for example, "age" rapidly to maximum strength; normally, others are artificially aged by a process termed precipitation heat treatment, which is a low-temperature thermal treatment.

It is in the interim period between quenching and aging that much forming can be done. This period may be prolonged by refrigerating the metal, which prevents or retards precipitation of the alloying particles. Formed parts are subject to a minimum of distortion in the low-temperature precipitation heat treatment that then follows. This general method is used extensively by aircraft manufacturers in stretch forming sections of considerable area.

Technical assistance in connection with aluminum alloy selection and fabrication techniques may be obtained through Kaiser Aluminum & Chemical Sales, Inc., 1924 Broadway, Oakland 12, California. Copies of a product information book on aluminum sheet and plate, containing comprehensive tables, may also be obtained on request.

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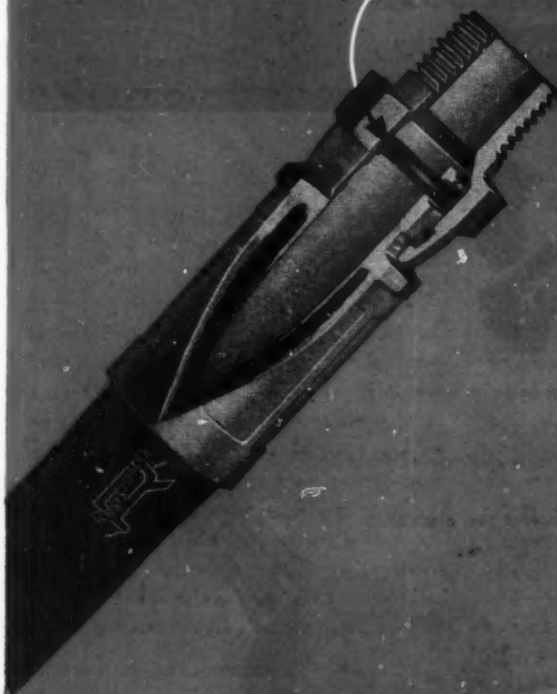

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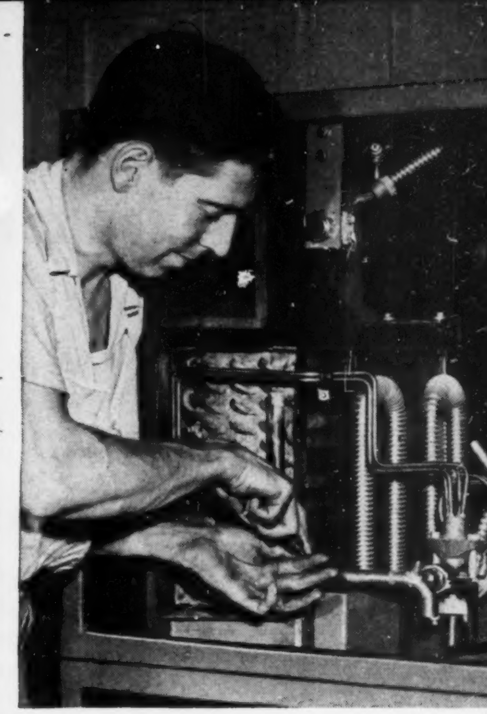

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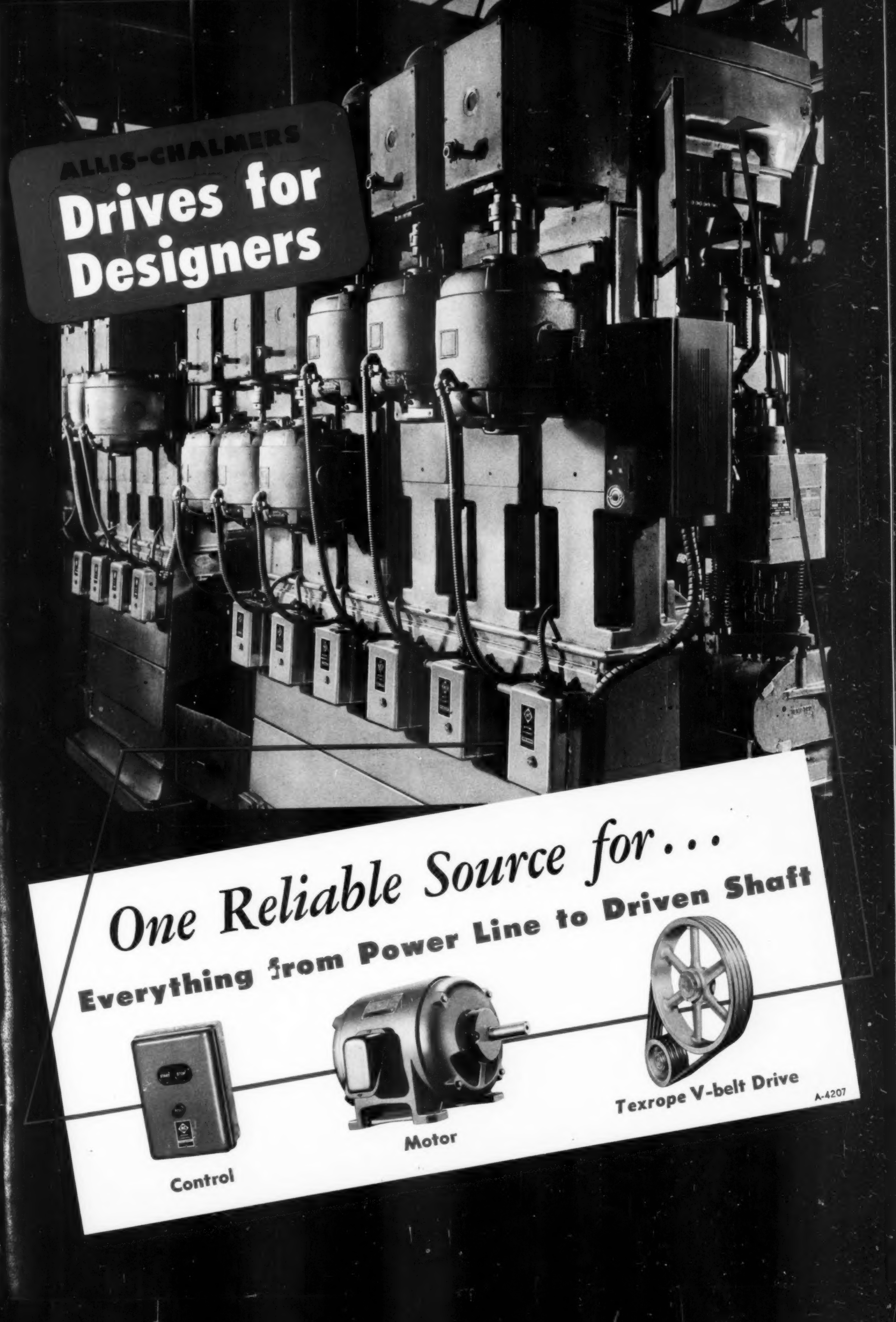

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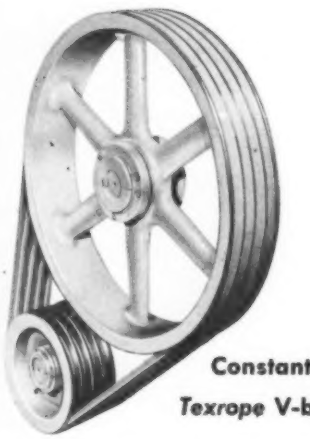

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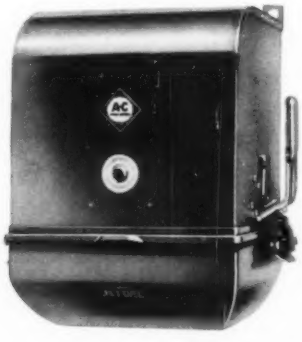

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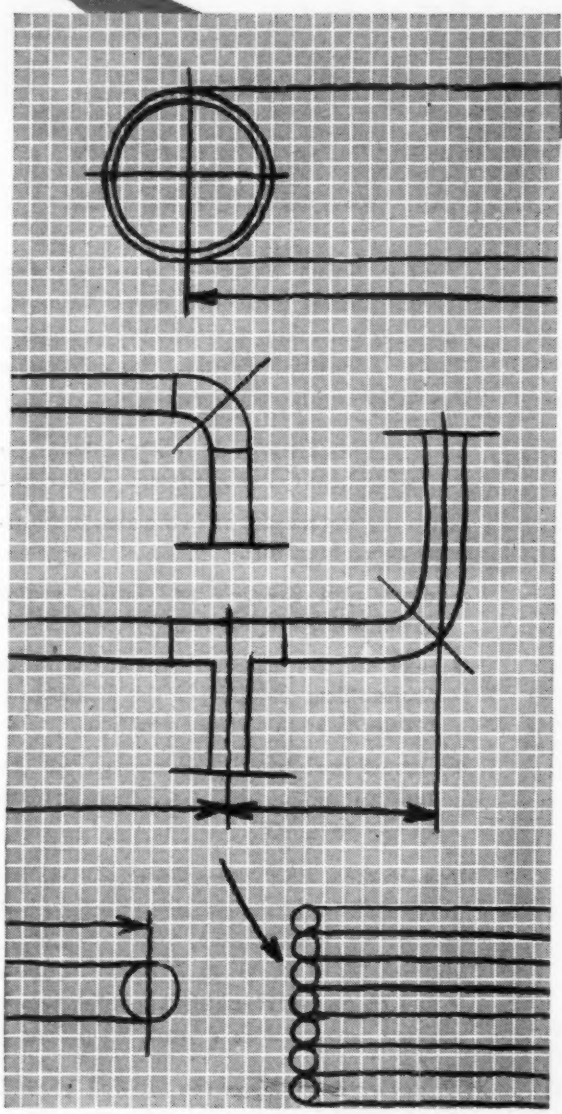

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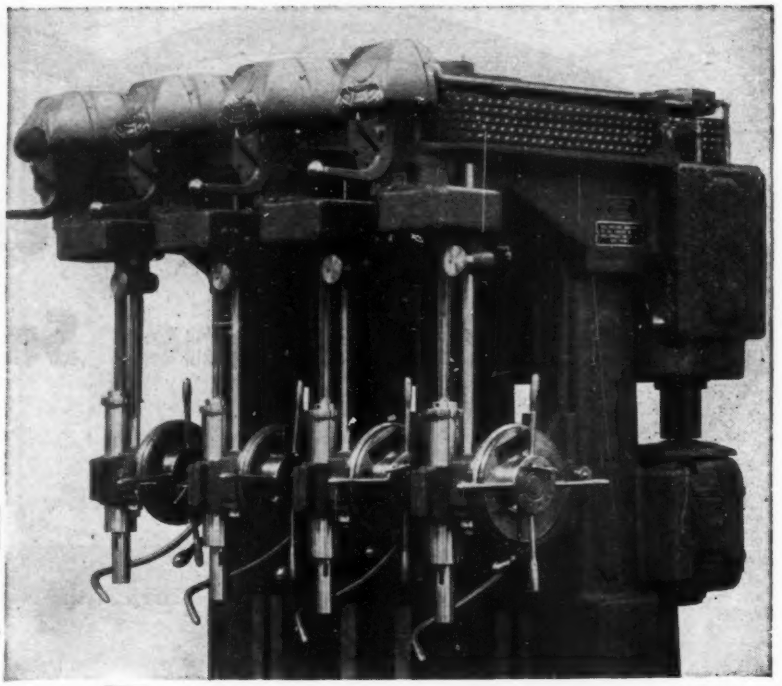

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- **Belts can be installed quickly without dismantling any part of the machine—outboard bearings are practical.**
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Use Veelos, the adjustable v-belt, for unlimited freedom in designing v-belt drives. Veelos is made in all standard widths: 00, 0, A, B, C, D and E, double V in 0, A and B. Available in three types: regular, oil-proof and static conducting.

Write for your copy of the fact-full VEELOS DATA BOOK . . . it's yours for the asking!

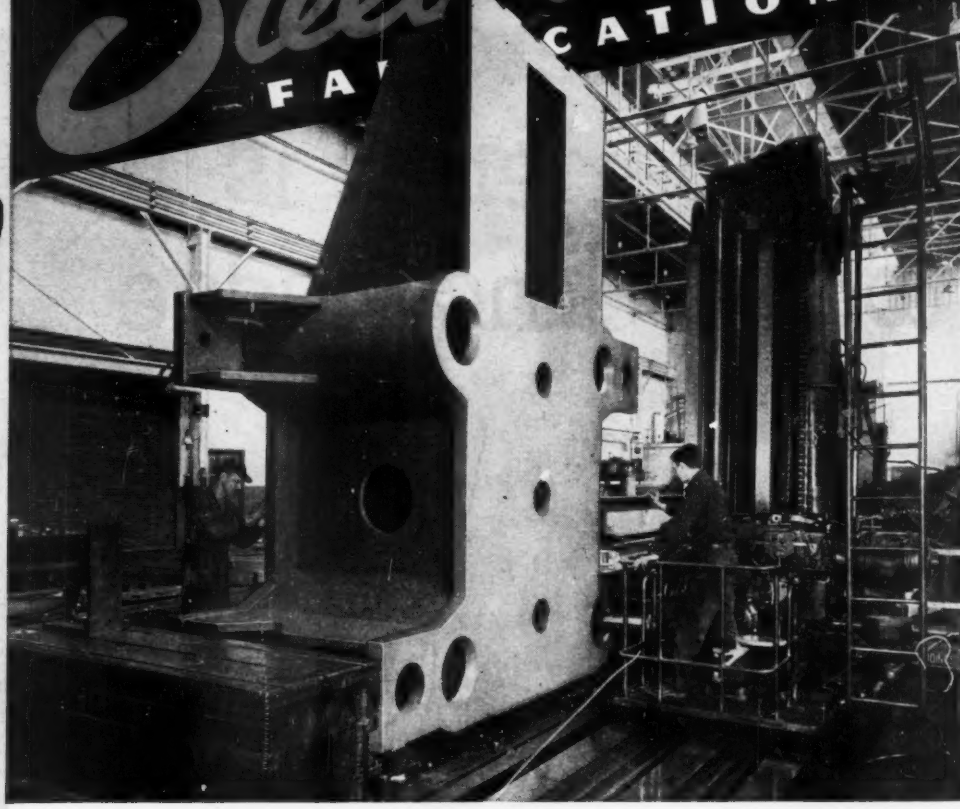
MANHEIM MANUFACTURING & BELTING COMPANY
606 MANBEL STREET
MANHEIM, PENNSYLVANIA

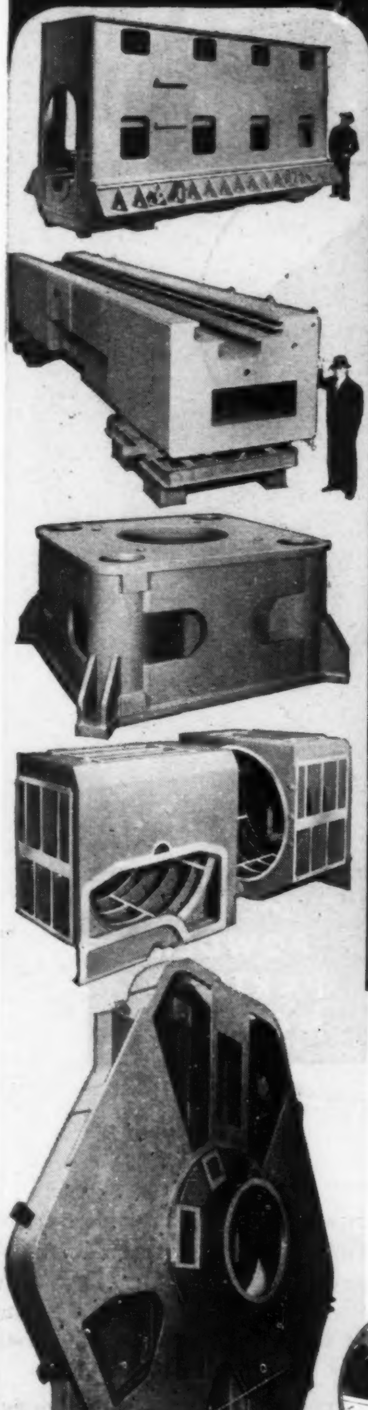
VEELOS
THE LINK
V-BELT

ADJUSTABLE TO ANY LENGTH . . . ADAPTABLE TO ANY DRIVE

Veelos is packaged on reels in 100-foot lengths. Sales engineers in principal cities; over 350 distributors throughout the country. Veelos is known as VEELINK outside the United States.

Steel-Weld FABRICATION





Use WELDED STEEL for Greater Strength with Less Weight!

The forty ton Press Platen above, and the parts and assemblies illustrated at the left, are typical of thousands of Steel-Weld Fabricated units produced and machined by Mahon for hundreds of manufacturers of heavy machines and other mechanical equipment. If parts of your product could be redesigned and produced to better advantage through Steel-Weld Fabrication, or, if you require a limited number of large heavy pieces in which pattern costs are a consideration, you can turn to The R. C. Mahon Company with complete confidence . . . personnel and facilities are available within the Mahon plant to do the complete job from drawing board to finished machining. You will find in the Mahon organization a unique source with complete ultramodern fabricating, machining and handling facilities to cope with any type of work regardless of size or weight . . . a source where skillful designing and advanced fabricating technique are supplemented by craftsmanship which assures a smoother, finer appearing job embodying every advantage of Steel-Weld Fabrication. See Mahon's Insert in Sweet's Product Design File, or write for further information.

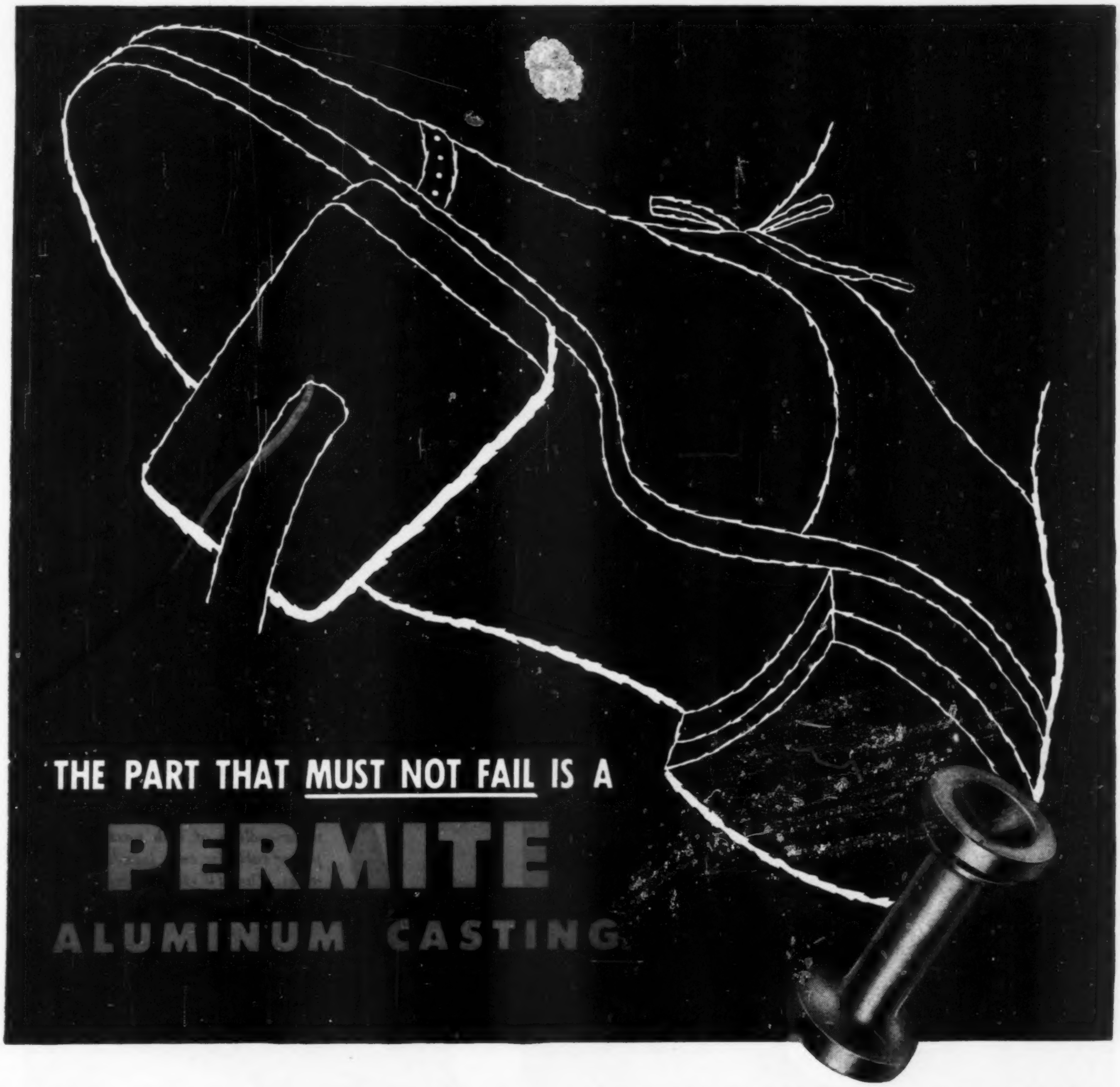
THE R. C. MAHON COMPANY

DETROIT 34, MICHIGAN

Engineers and Fabricators of Steel in Any Form for Any Purpose

MAHON

not in the blueprint..
A great many who send blue prints to the Bunting Brass & Bronze Company have in mind things not shown by the drawings. They know that in the event of an unforeseen happening of any kind they can depend on the Bunting organization for consideration, cooperation and prompt, specific help. Ask any Bunting customer.
The sleeve bearing is not complex in structure; it conforms readily to design requirements; it is easily installed. Cast Bearing Bronze is the most adaptable of all sleeve bearing materials; it possesses excellent anti-friction properties. With proper film lubrication, its coefficient of friction is as low as can be obtained with any other bearing type. A successful Bunting Bronze bearing installation is readily attainable. We ask the opportunity to work with you, and to quote on your requirements.
Bunting®
BRONZE BEARINGS • PRECISION BRONZE BARS • BUSHINGS
THE BUNTING BRASS & BRONZE COMPANY • TOLEDO 1, OHIO • BRANCHES IN PRINCIPAL CITIES



It's a fact! In 90% of all cars, trucks, and buses on the highways today, the master cylinder brake piston is a Permite Aluminum Alloy Casting, made by the permanent mold process by Aluminum Industries, Inc.

Yes, for the one vital part that *must not fail*, automotive manufacturers turn to Permite, because ever since the early pioneering days of the use of aluminum, Permite Aluminum Castings have enjoyed a reputation for highest quality.

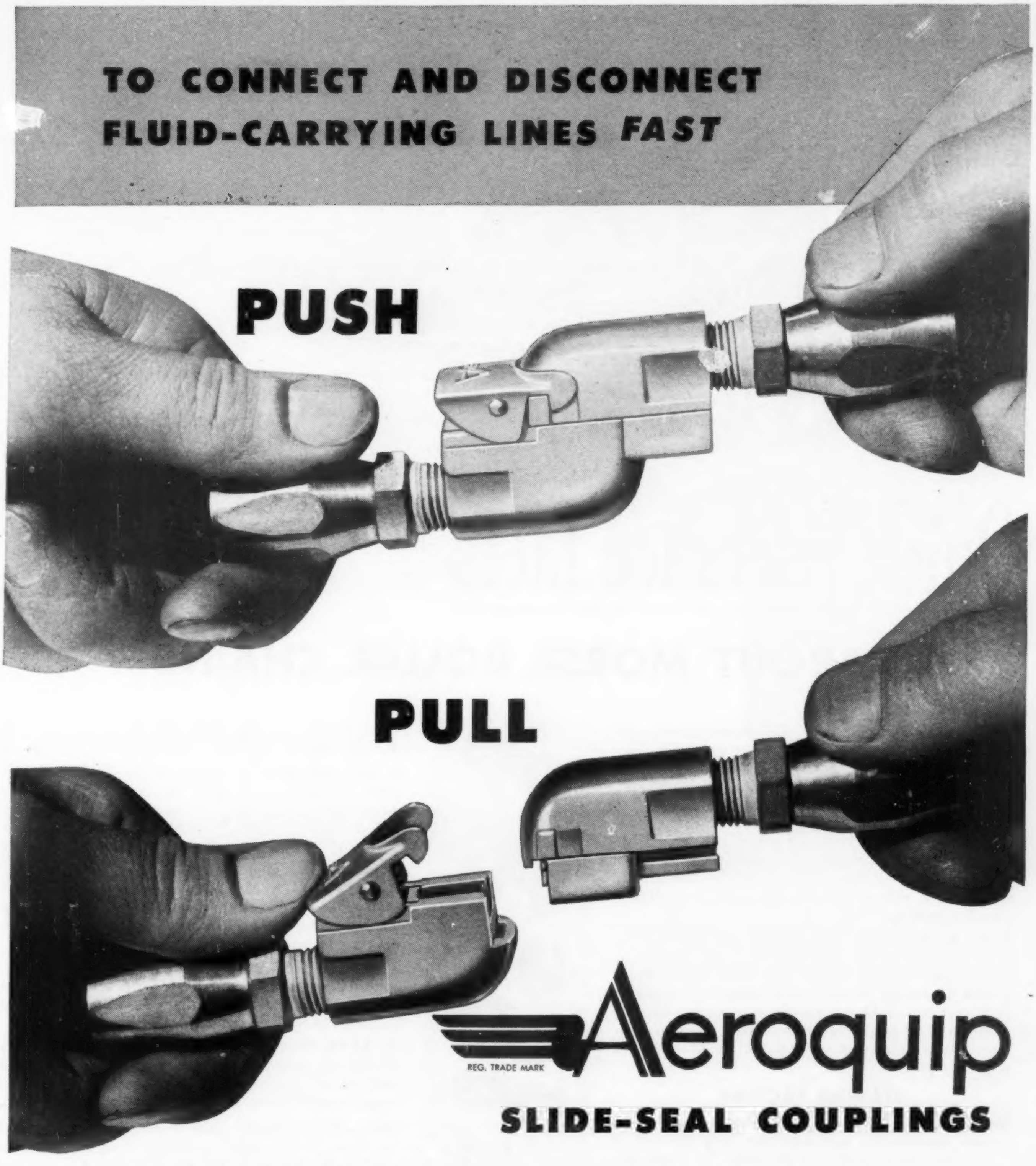
Component parts made of Permite Aluminum Castings, permanent mold and sand cast, are contributing to faster machining, speedier production, lower costs in all divisions of the metalworking industry. If you want castings of assured structural strength, accurate in design and dimension, free of hidden defects, castings that will sharply reduce your reject rate, consult with Permite engineers on your requirements. Send blue prints for recommendations and quotations.

ALUMINUM INDUSTRIES, INC. • CINCINNATI 25, OHIO

Detroit: 809 New Center Bldg. • New York: 9 Rockefeller Plaza • Chicago: 64 E. Jackson Blvd.

PERMITE **Aluminum Castings**

ALUMINUM PERMANENT MOLD and SAND CASTINGS . . . HARDENED, GROUND and FORGED STEEL PARTS



SLIDE-SEAL COUPLINGS

This Aeroquip development saves you time and money!

LOW COST . . . Aeroquip Slide-Seal Couplings connect and disconnect fluid-carrying lines instantly assuring full flow of fluids when connected, and perfect seal of each half when disconnected.

LIGHTWEIGHT AND COMPACT . . . They are ideal for use in confined areas and may be used to advantage in a wide variety of applications.

SIMPLE DESIGN . . . Only four working parts and two "O" rings assure positive performance and foolproof operation.

ADAPTABLE for use with many fluids including hydraulic fluids, hot oil, crude and fuel oils, anti-freeze solutions, gasoline, Diesel fuels, air, water, and other fluids.

Descriptive literature is available . . . please write.

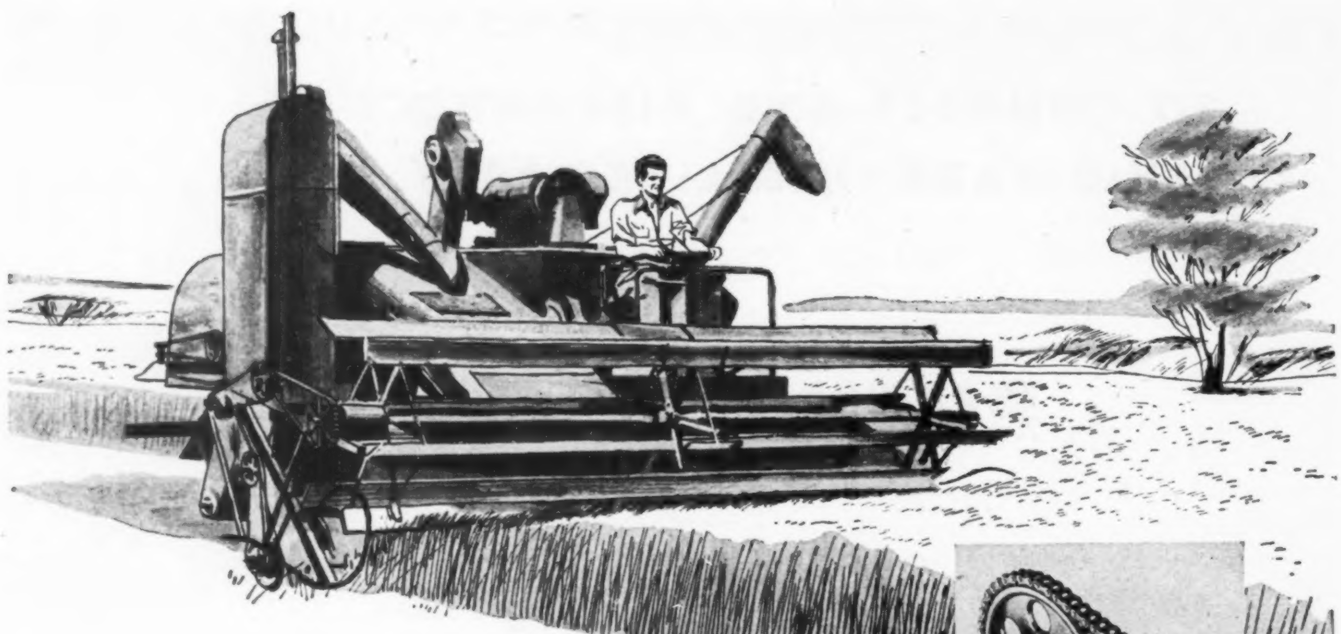
AEROQUIP CORPORATION, JACKSON, MICHIGAN

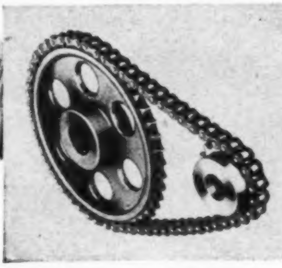
SALES OFFICES: Burbank, Calif. • Dayton, Ohio • Hagerstown, Md. • High Point, N. C. • Miami Springs, Fla. • Minneapolis, Minn. • Portland, Ore. • Wichita, Kan.

IN CANADA: Prenco Progress and Engineering Corporation Ltd., Toronto

IN ENGLAND: Super Oil Seals & Gaskets, Ltd., Birmingham

AEROQUIP PRODUCTS ARE FULLY PROTECTED BY PATENTS IN U.S.A. AND ABROAD





Quick refresher facts

ABOUT MORSE ROLLER CHAINS

You gain design flexibility, hold power-transmission component costs to a minimum, add new operating stamina to your products when Morse Roller Chains and Sprockets are at work for you.

DESIGN, ECONOMY FACTORS

Morse Roller Chains transmit power positively with less than 1% friction loss. They operate with maximum efficiency on long or short centers, mesh with sprockets on either side of chain to drive multiple-sprocket hookup.

Morse Roller Chains bring about major production economies in your plant. They run efficiently and well despite shaft end play, unparallel shafts. They permit increased drive center-distance tolerances. And because of Morse precision mass production methods, the complete drives come to you produced under the most economical conditions competitively priced.

STAMINA FACTORS

Morse Roller Chains will help you build even greater stamina into your machines, help you reduce downtime and maintenance problems for your customers. Here are a few reasons why:

1. PINS. Heat-treated special high-nickel, fine-grain alloy steel pins have hard case for wear resistance, tough inner core for high strength. Finish ground to close limits.

2. PIN LINKS. In Morse Roller Chains, heavy press fit holds pins immobile in pin link plates. Accuracy of assembly and close finish tolerance of pin and plate components assures proper clearances between pins and bushings for penetration by lubricant.

3. BUSHINGS. Morse bushings are case hardened and curled for maximum wear. Smooth inner bearing surfaces and true roundness help eliminate pin scoring, increase joint life, reduce elongation.

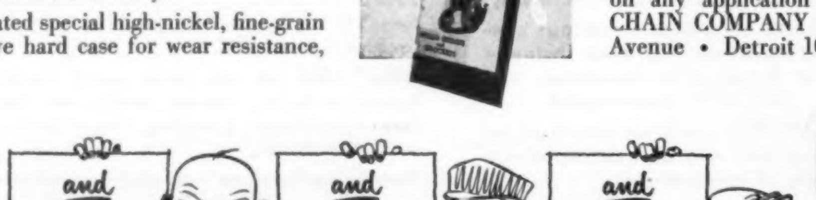
4. ROLLERS. Rollers are shot-peened for maximum wear and resistance to shock. Rollers are processed from special alloy steel. Close-tolerance surfaces provide even load distribution.

5. LINK PLATES. Morse link plates are specially treated to obtain maximum structural strength and endurance qualities necessary in highly stressed tension parts. Holes are accurately pierced and sized.

Morse Roller Chains, with corresponding sprockets in Types A, B, and C available from ⅜" pitch to 2½" pitch. Write for Catalog C51-50, or for details on any application you have in mind. MORSE CHAIN COMPANY • Dept. 566 • 7601 Central Avenue • Detroit 10, Michigan.

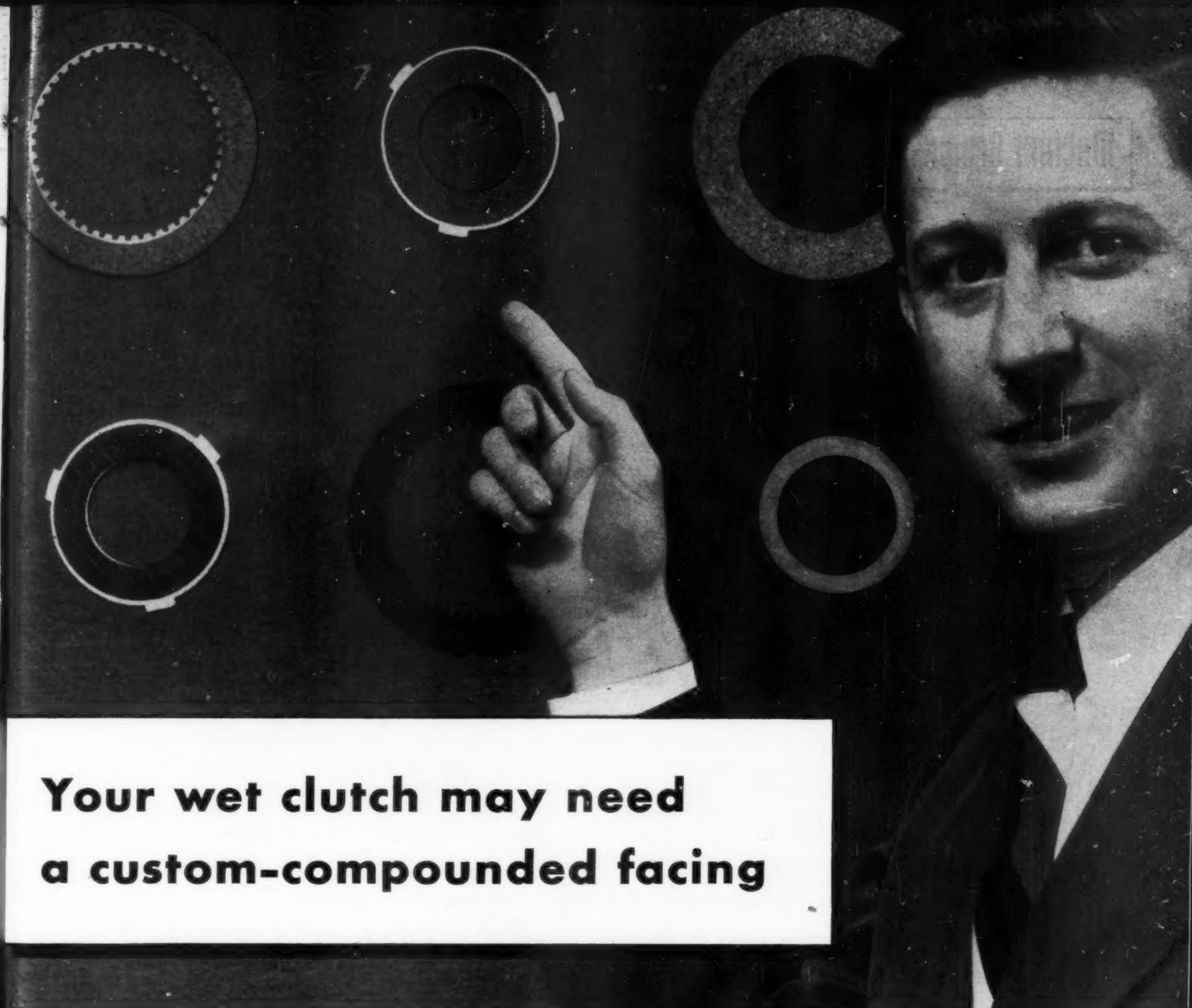
M-PT

MORSE *means* POWER TRANSMISSION



MORSE

MECHANICAL POWER TRANSMISSION PRODUCTS



Your wet clutch may need a custom-compounded facing

A well-designed clutch may lose a lot of its inherent performance if the torque-engagement characteristics of its facing don't precisely fit its mechanical characteristics. That's why your wet clutch may need a facing material that's compounded to supply a particular combination of properties.

Of course, one of Armstrong's many standard friction materials may be exactly what you need. But if you need something special, Armstrong can supply it with a minimum of time and effort on your part.

You merely give us the pertinent data on your clutch design—which you can do easily on one of our Clutch Data Sheets. Then we will give you a selection of facings that seem most suitable, either in cork or combinations of cork and rubber, or cork and fiber.

If your field tests show that none of these materials gives you the performance you're looking for, and very often one will, then your test data can be used by Armstrong compounders either to modify an existing compound or create an entirely new one. Here's how they do it.

All Armstrong's Friction Materials are performance-tested under standard, controlled conditions on special, Armstrong-built test equipment. Your test data on the most promising of the facings are compared with the laboratory data on that same material. This comparison shows how it may need to be modified or which other Armstrong material may already offer the properties you need.

If a compound modification or a new compound is called for, Armstrong compounders ordinarily can come up with it quickly because their studies already have established what the effects are of many variations in formulation or operating condition. This has a double advantage for you. You can get a suitable facing with a minimum of experimental time and expense. You also have assurance that the material you select will do the best possible job in your particular clutch.

If you're working on a new clutch design now, or would like to improve the performance of an existing unit, write for a copy of our Clutch Facing Data Sheet. We'll be glad to study the data you submit, suggest suitable materials, and supply samples for experimental use. Write Armstrong Cork Company, Industrial Division, 7212 Dean Street, Lancaster, Pennsylvania.

ARMSTRONG'S FRICTION MATERIALS

MACHINE DESIGN Topics ...IN ENGINEERING AND RESEARCH

Rotating Wheel Recovers Heat Lost Through Ventilation

Operating on a new heat recovery method (heat regeneration), a new device is said to retain up to 96 per cent of heat generally lost through exhaust ventilation. Permitting almost complete transfer of temperatures between exhaust and fresh make-up air, the Therm-O-Wheel consists of a slowly revolving wheel containing metallic wool. Exhaust air is blown through the upper half of the rotating wheel, while incoming fresh air is blown through the lower half. Heat picked up by the metal wool in the exhaust air is thus transferred to incoming air. The wheel is said to operate efficiently at almost any temperature differential.

Apricot Pits, Corncobs and Dry Ice

More than 80,000 apricots a month give up their pits to smooth plastic parts for insulating components at the Square D Co. The pits, pulverized to the consistency of bird seed, provide a relatively soft material for "sandblasting" rough edges of plastic parts. Apricot pits are not the only unusual medium so used; walnut shells, maize and ground-up corn cobs are used for soft metals and plastic; carbon is blasted out of engine cylinders with rice; and pulverized dry ice is used to deflash rubber parts.

New Naval Acoustic Laboratory Studies Underwater Sound

Underwater sound-pickup devices will be studied at a new floating laboratory on a Patuxent River reservoir north of the Capitol. Frequencies ranging from subaudio to high ultrasonic can be used, and sound projectors, hydrophones and transducers will be tested. Testing is performed from a covered barge with an enclosed well through which experimental devices can be lowered to the bottom of the reservoir.

Nylon Fuel Cells Replace Integral Aircraft Wing Tanks

Fuel cells in a Lockheed Lodestar owned by Goodyear Tire and Rubber Co. will be made of nylon. Installed by AiResearch Aviation Service Co., the fuel cells will replace an integral tank system in which fuel is contained by the skin and cross-members of the wings, thus eliminating the need for periodic resealing along joints.

"Pocket-Size" X-Ray Sources Developed

Self-contained X-ray sources, varying in size from pellets to larger blocks or sheets, may soon be a commercial reality. Radiation emitted is essentially the same as that emitted by bulky X-ray equipment, but of lower intensity. The source, which operates on what is known as the Bremsstrahlung principle, consists of a radioactive material surrounded by a layer of heavy metal. Unlike ordinary radioisotopes, the source produces only desirable X-rays, and the device can be designed to remain active for almost any time period desired.

Indexing System Permits Significant Time and Space Savings

Saving up to 90 per cent in present catalog space and up to 75 per cent in cataloging time, a new indexing system developed by the Armed Services Technical Information Agency is said to be a significant step in in-

ANOTHER NEW FAFNIR...

Ball Bearing

"Economy Package"

Type PB Pressed Steel Pillow Block

The new Type PB Series Pillow Block is developed to fill a specific need for a high-quality, precision ball bearing unit at the lowest possible cost. It is made for *light duty* applications where a low-cost "packaged" unit offers definite manufacturing and servicing advantages . . . on agricultural equipment, conveyors, light-duty fans and similar types of machinery involving the transmission of power.

The Type PB Series Pillow Block opens up new opportunities to add extra product sales features and to lower maintenance costs . . . another example of the Fafnir "attitude and aptitude". Send for new, descriptive bulletin. The Fafnir Bearing Company, New Britain, Conn.

OUTSTANDING FEATURES

- Low Cost
- Precision Ball Bearing . . . Fafnir Wide Inner Ring Bearing* with Self-locking Collar and efficient Plya-Seals.
- Two-piece separable pressed-steel housing.
- Self-aligning . . . unrestricted in all directions at assembly.
- Occupies less space than other pillow blocks.
- Light weight . . . combined with ample strength.
- Only two bolt holes required in assembly.
- Ample capacity for radial, thrust or combined loads.
- Pre-lubricated with long-life grease.
- Available for shaft sizes ½" to 1¼" inclusive.

*Easiest bearing of all to install. Counter-bored, eccentric, mated cam construction of collar and inner ring assures positive locking action without set screws, lock nuts or adapters.

FAFNIR
BALL BEARINGS
MOST COMPLETE LINE IN AMERICA

formation storage and retrieval by machine methods. In conventional catalogs or indexes, information on aircooled aircraft guns would require six entries to be complete: (1) aircraft guns, aircooled, (2) guns, aircraft-aircooled, (3) aircooled guns, aircraft, (4) guns, aircooled, aircraft, (5) aircraft-aircooled guns, and (6) aircooled-aircraft guns. With the Uniterm System of Co-ordinate Indexing, three cards would be used; all information on "guns" would be recorded on one and, similarly, information for "aircraft" and "aircooled" on the others. Numbers (used to identify references) common to all three cards would indicate information on aircooled aircraft guns, with complete assurance that all references on the subject had been located.

Artificial Eyes—Made with Magnets

An unusual use for permanent magnets is in an artificial eye developed recently by a New York doctor. As reported by Indiana Steel Products Co., an Alnico magnet is imbedded in an implant which is connected to the muscles which move the eye. The implant must then be covered with layers of tissue and completely buried. A custom-fitted plastic artificial eye, essentially a curved shell with a very thin Alnico magnet, is made. The natural and artificial eyes can then move in unison, due to the attraction between the magnets.

Electromagnetic Pumps Used for Automatic Ladling

A successful application of electromagnetic pumps, in which molten metal is pumped by magnetic force only, is permanent-mold casting of automotive pistons. A moving line of molds is fed from an orifice which can discharge from ½ to 5 pounds of aluminum alloy per second depending on orifice size. A silicon-carbide tube concentrates the circulating forces which lift and discharge the metal through the orifice; a resistance heater close to the nozzle prevents freezing. With automatic cycle control by means of a photoelectric cell, pistons can be cast at a rate of 2000 per 8½ hour shift.

First Objects to be Chrome-Plated—Doorknobs

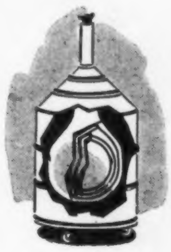
A pair of ordinary doorknobs at Columbia University were "retired" recently when it was discovered that they were the first objects ever chrome-plated by a successful commercial process. At the death of Dr. Colin G. Fink, inventor of the process, search was instituted last month for mementoes of his early work. The doorknobs, which were plated on April 3, 1924, were located on the doors of one of the University's rooms and will be retired to the Chandler Chemical Museum.

Electronic Device Speeds Up Inventory Control

A new electronic device, called "Speed Tally" by its builder, Remington Rand Inc., is expected to revolutionize inventory control. The latest mathematical wizard, as developed for a Chicago business, maintains running tallies of inventory items in over 13,000 classifications. Ten keyboard devices that look like adding machines provide the input. To enter an item, the quantity and a code number is punched. Each catalog number is assigned a location on a magnetic memory storage drum; when a code figure is punched the "address" of the classification is located and the existing total is picked off the drum, added to the input, and the total returned to the drum. The entire operation requires approximately 0.4-second, and scanning of all keyboards requires less than 5 seconds.

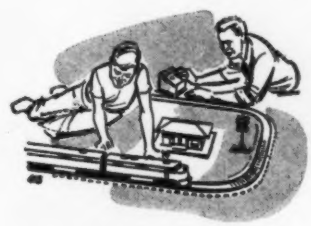
ANACONDA®
METALS AT WORK

Copper "package" for the coldest products on earth



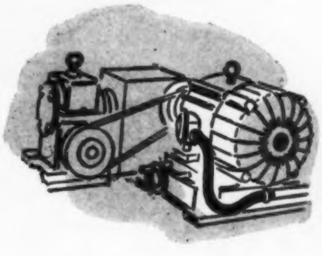
This container for storing and transporting liquefied helium (—452 F.) or hydrogen (—421 F.) is made by Superior Air Products Co. of Newark, N. J. The secret of keeping heat out to reduce evaporation is four concentric copper spheres separated by a high-vacuum space, a nitrogen protecting bath, and a second high-vacuum space — the entire unit enclosed in a stainless steel casing. Copper contributes ductility and malleability for spinning into hemispheres, unexcelled soldering properties for joining into spheres, and a mirror finish to reduce radiation losses. Need we tell you that it's made of ANACONDA Copper?

This "iron horse" runs on brass



The "High Iron" we're referring to is the HOBBYLINE HO-GAGE model railroad track which John A. English & Co. of Morrisville, Pa., makes of FORMBRITE** strip. This fine-grain, yet ductile forming brass produces a track so strong and stiff it will support an adult's weight. In products where finishing costs count, users find that FORMBRITE usually needs only a color buff to produce a high finish.

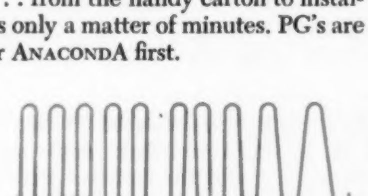
Flexible conduit enjoys a splash in a bath



The real trick is to make an electrical conduit that's flexible *and* liquid-tight *and* approved by Underwriters' Laboratories for such applications. SEALTITE* Type UA does all three. Made with a tough synthetic jacket extruded over a flexible steel core, it's shown here protecting wiring to motor in an auto laundry. It ignores not only splashing, but also oil and grease, and resists abrasion. Want more facts?

We put an end to "the bends"

We don't mean the kind deep-sea divers suffer from, but the kind heating contractors had to put up with when they installed radiant panel heating systems. Heretofore, many contractors formed the panel grids on the job by hand-bending them from coils of tubing—a time-consuming, laborious, back-breaking job. Now PG's** have ended all that. They are the new, compactly packaged, accurately *preformed* ANACONDA Copper Tube Panel Grids . . . from the handy carton to installation is only a matter of minutes. PG's are another ANACONDA first.



A service for you

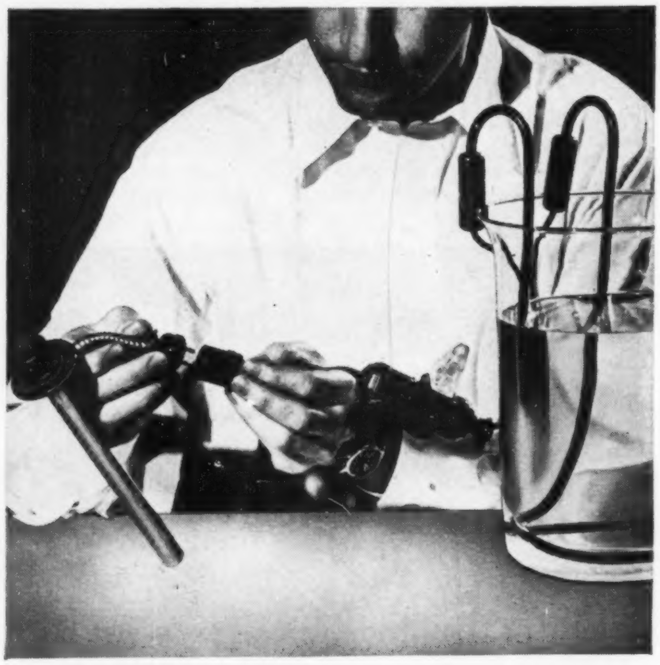
Our Technical Department has a range of experience that covers the entire field of copper and copper-alloy applications in industry. If you have a problem of metal selection, we are at your service. *The American Brass Company, Waterbury 20, Conn. In Canada: Anaconda American Brass Ltd., New Toronto, Ont.*

*Trademark
**Reg. U.S. Pat. Off.
5393

ANACONDA® ***the name to remember in* COPPER — BRASS — BRONZE**

How would YOU solve these two problems?

1. FLINTKOTE INSULATED SIDING is saturated with hot asphalt by a continuous conveyor process through a dip tank. Fenwal THERMOSWITCH® Thermostats, connected to pilot lights, warn when asphalt varies from safe operating level. The totally enclosed mechanism of the THERMOSWITCH units is not affected by the high heat (420°F) or viscosity of the liquid.



2. LABORATORY WORK SAVED. Let one Fenwal All-Purpose Laboratory THERMOSWITCH thermostat do the work of many. This unit has a quick-disconnect plug — goes readily from job to job. It provides close temperature control for solids, liquids and gases. Stainless steel shell, resistant to corrosion, shock and vibration.

3. YOU, TOO, MAY BENEFIT from a Fenwal THERMOSWITCH Control.* Compact, highly resistant to shock and vibration, Fenwal THERMOSWITCH units have solved many types of temperature control problems throughout industry. The external, single-metal shell expands or contracts *instantly* to temperature changes, making or breaking electrical contacts.

*Cartridge Type Illustrated

4. SEND FOR THIS NEW CATALOG for complete explanation of the unique THERMOSWITCH unit. Also ask for more detailed, illustrated discussions of the problems above. Fenwal engineers will be glad to help you solve your temperature control problems involving heat, humidity, radiant heat, pressure and other variables. Write **Fenwal Incorporated, 1912 Pleasant St., Ashland, Massachusetts.**

THERMOSWITCH®

Fenwal

Electric Temperature Control and Detection Devices

SENSITIVE . . . *but only to heat*

Data supplied by the Radio Condenser Company, Camden, N. J.

IT'S MADE OF BERYLCO BERYLLIUM COPPER

Why beryllium copper is specified more and more for electronic equipment is illustrated by this wiper contact for a variable condenser used in Signal Corps combat transmitting and receiving equipment. It is made of quarter hard Berylco 165 strip, .014'' thick, cadmium plated. The contact buttons are silver.

Electrical conductivity here is a must, of course. But even more important is the degree of contact the part makes with the shaft. Too little pressure will make a poor electrical contact; too great a pressure will excessively load the shaft. In production, the wiper is tested with a combination leaf spring gage and electric continuity checker to register between 40 and 70 grams of pressure.

Once this adjustment is made, the Berylco contact will exert a constant pressure, come what may. In fact, it will remain in adjustment even after hand-forming during the checking operation.

It is just this ability of Berylco to solve conflicting mechanical, electrical and service requirements that makes it valuable to designers. Here, in one alloy, are combined such properties as strength, conductivity, corrosion resistance and elasticity—a combination no other alloy gives you.

If you would like to include beryllium copper in your plans for the future, we invite you to share the know-how of the world's largest producer of this unique alloy. For engineering help or sample material, call or write any of the offices listed below.

INVALUABLE

This 16-page booklet, called APPLICATIONS UNLIMITED, contains nearly 40 case histories — shows how Berylco has solved many troublesome design and engineering problems. Send for your free copy today.

TOMORROW'S PRODUCTS ARE PLANNED TODAY... WITH BERYLCO BERYLLIUM COPPER

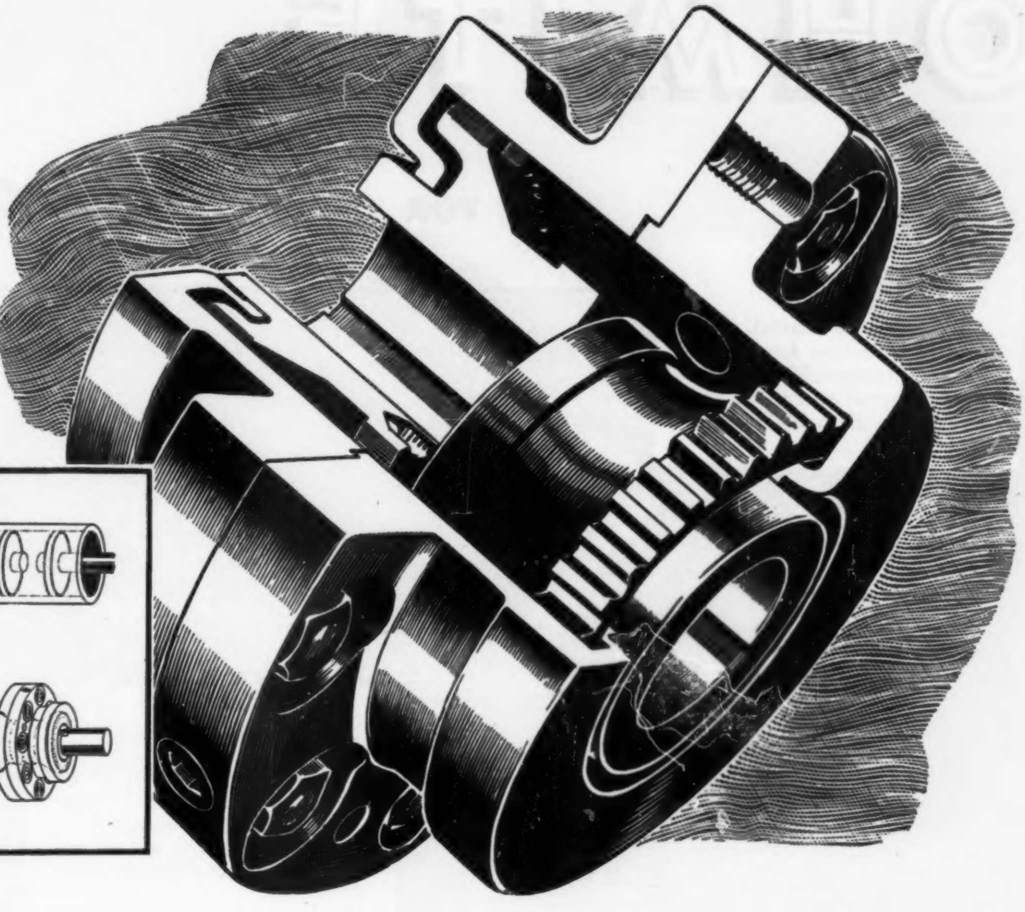
BERYLCO THE BERYLLIUM CORPORATION

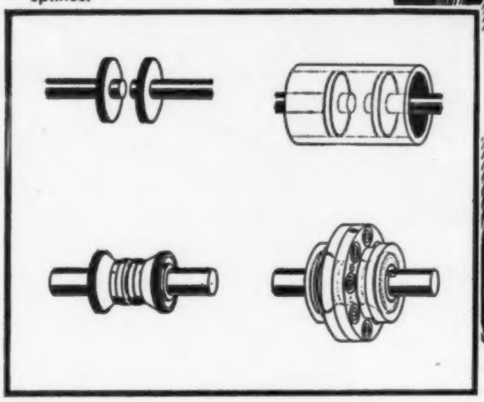
DEPT. 3L, READING 1. PENNSYLVANIA

New York • Springfield, Mass. • Rochester, N. Y. • Philadelphia • Cleveland • Dayton • Detroit • Chicago • Minneapolis • San Francisco • Los Angeles

Representatives in principal world-trade centers

FAST'S CONSTRUCTION is simple as A-B-C! A splined hub on each shaft end. A sleeve with internal splines to mesh. Oil in the sleeve, kept between the splines by centrifugal force, carries the load between the splines.





For free engineering service that saves dollars and down-time . . .

FAST'S Couplings are FIRST!

When you specify Fast's Couplings, you get the benefit of Koppers *free engineering service* . . . your assurance of the right coupling for any job and the right solutions to tough coupling problems.

FOR EXAMPLE: "Get the drawings and send them down to Koppers" is the standard answer in one plant whenever a tough coupling application arises!

FOR EXAMPLE: After studying a list of spares for stand-by units made up by one customer, a Koppers sales engineer eliminated duplications . . . resulting in a smaller order for him but a *much smaller inventory* for the customer!

Add the *rugged construction* of Fast's Couplings . . . with their original design maintained without basic change or sacrifice in size or materials. Add their lowest cost per year . . . their life expectancy guarantee. *Fast's usually outlast the equipment they connect.* Result: it will pay you to write today for full details on how Fast's Couplings and Koppers Engineering Service can help you get uninterrupted power transmission! Send for free catalog to: KOPPERS COMPANY, INC., *Fast's Coupling Dept.,* 342 Scott Street, Baltimore 3, Md.

THE ORIGINAL

KOPPERS

Fast's Couplings

METAL PRODUCTS DIVISION • KOPPERS COMPANY, INC. • BALTIMORE, MD. This Koppers Division also supplies industry with American Hammered Industrial Piston and Sealing Rings, Koppers-Elex Electrostatic Precipitators, Aeromaster Fans and Gas Apparatus.

Engineered Products Sold with Service

KOPPERS COMPANY, INC., ***Fast's Coupling Dept.,*** **342 Scott St., Baltimore 3, Md.**

Gentlemen: Send me a Fast's Catalog giving detailed descriptions, engineering drawings, capacity tables and photographs.

Name..

Company..

Address..

City.. Zone .... State ....

OHMITE®
FOR Dependability
RHEOSTATS
All-ceramic and metal, close control rheostats for unsurpassed dependability and smoothness of operation. Ten stock sizes, 25 to 1,000 watts.
TAP SWITCHES
Five compact models, 10 to 100 amperes, AC, up to 12 taps. All-ceramic and metal construction. Silver-to-silver contacts, with self-cleaning rotor.
RESISTORS
A wide range of dependable, fixed, adjustable, tapped, and non-inductive, power wire-wound resistors. Also a wide range of precision resistors.
R. F. CHOKES
Single layer R. F. plate chokes and power line chokes, on steatite or plastic cores. Protected by a special moisture-resistant coating.
OHMITE
Be Right with OHMITE
OHMITE MANUFACTURING COMPANY
3618 HOWARD ST., SKOKIE, ILLINOIS (Suburb of Chicago)
WRITE on company letterhead for Catalog and Engineering Manual No. 40

WANTED...

Engineer with writing interest and ability

MACHINE DESIGN needs an assistant editor with background in mechanical engineering.

Principal duties are to seek out and organize information from various sources into articles of practical value to design engineers.

Essential qualifications are enthusiasm for writing and a strong creative urge to be of service to our readers through the development and dissemination of the latest engineering knowledge.

If you are interested and believe you are qualified, please write giving full details about your experience to the editor, MACHINE DESIGN, Penton Building, Cleveland 13, Ohio.

what's your problem?

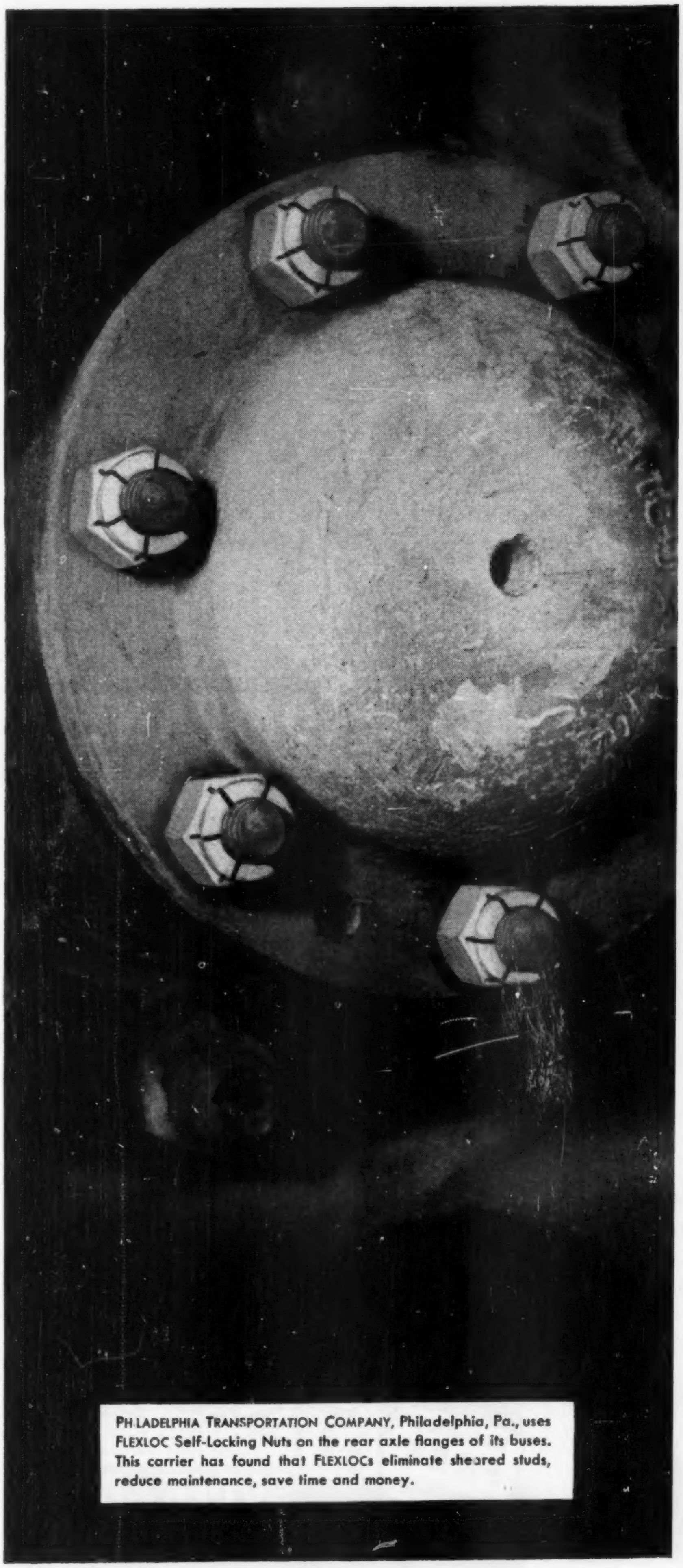
Parts? Finishes? Components? Materials?
Maybe one of the advertisements in this issue of MACHINE DESIGN has the answer to your current mental whiplash.

Or maybe the answer is suggested but you want to find out for sure. Why not spend a few seconds right now and get it over with?

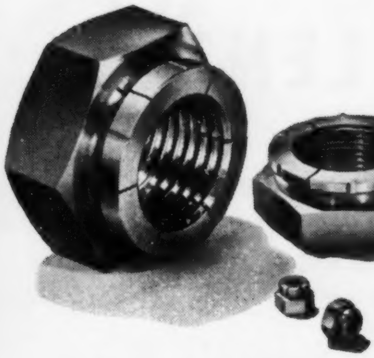
Fill out one of the advertising inquiry cards (always *yellow*) and shoot it to us. No letters or postage necessary.

We'll send your inquiry immediately to the advertiser and he will then reply directly to you.

USE THE YELLOW CARDS ON PAGE 39



PHILADELPHIA TRANSPORTATION COMPANY, Philadelphia, Pa., uses FLEXLOC Self-Locking Nuts on the rear axle flanges of its buses. This carrier has found that FLEXLOCs eliminate sheared studs, reduce maintenance, save time and money.



How FLEXLOC locknuts work

FLEXLOCs lock and stay put on a threaded member regardless of the vibration encountered. Here's how they work. The slotted top or locking section is divided into six equal, flexible segments, closed in to make the inside diameter of the nut smaller than that of the companion bolt. When the FLEXLOC is applied, these are expanded by the bolt. The spring tension of the resilient segments locks the nut securely at any desired position on the bolt once the locking threads are fully engaged.

FLEXLOCs can be used over and over again. When expanded by the bolt, the locking section remains within the elastic limit of the metal. This permits the locking segments to return to their normal position, ready for reapplication to the bolt.

FLEXLOCs are one piece, all metal—nothing to assemble, come apart, lose or forget. They can be delivered in any quantity in a wide range of sizes. Stocks are carried by industrial distributors everywhere. Write for literature and samples. SPS, Jenkintown 18, Pa.

FLEXLOC®

LOCKNUT DIVISION

SPS

JENKINTOWN PENNSYLVANIA

Our Fiftieth Year : A START FOR THE FUTURE

HERE'S THE STRONGEST BRONZE

HY-TEN-SL
A Bronze as Strong as Nickel Steel

TRADE-MARK

Grade of HY-TEN-SL	1AA	1A	1	2	3	4
TENSION						
Ultimate tensile strength lbs. per sq. in.		(Minimums)				
Sand Cast	125,000	115,000	108,000	100,000	90,000	85,000
Rolled 1" and under	....	120,000	110,000	105,000	95,000	85,000
Rolled or Forged, over 1"	....	115,000	108,000	100,000	90,000	85,000
Yield Point (divider method)						
Sand Cast	95,000	75,000	60,000	55,000	45,000	40,000
Rolled 1" and under	....	75,000	65,000	60,000	50,000	45,000
Rolled or Forged, over 1"	....	75,000	65,000	60,000	50,000	45,000
Elongation in 2"—%						
Sand Cast	10	12	14	15	20	25
Forged or Rolled	....	12	13		15	20
Reduction in area—%						
Sand Cast	....	12	14	15	20	25
Forged or Rolled	....	12	13		15	20
COMPRESSION						
Yield Point— lbs. per sq. in.	70,000	65,000	58,000	50,000	40,000	35,000
Permanent Set at 100,000 lbs. per sq. in. (max.)	.015	.020	.030	.050	.080	.125
Brinell hardness, No.	250	240	220	200	175	150

List of U. S. Government Specifications Covering HY-TEN-SL Bronze

ARMY
Forgings, Rolled Rod QQ-B-721 Class B
Castings...QQ-B-726 Class B and C

NAVY
Forgings and Rolled Rod 46B15d Class B
Castings, Aluminum-manganese Bronze 46B29

WATERTOWN and OTHER ARMY ARSENALS
Castings WXS-5
Forgings and Rolled Rod......WXS-2

NAVAL GUN FACTORY
Washington Navy Yard
Castings—Trunnion Metal......O.S.550

AIRCRAFT
See Navy and Army (above).

Send for 46 Page "Text Book"

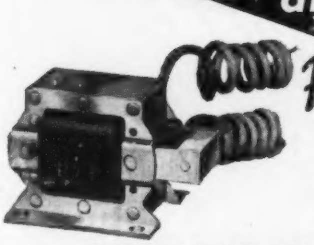
AMERICAN MANGANESE BRONZE COMPANY

4715 RHAWN ST., PHILADELPHIA 34, PA.

Established 1909

DORMEYER

Machine Tool SOLENOIDS are.. *Precision-Built* to LAST and LAST



Endurance is built right into each Dormeyer Solenoid. Performance records show that 17,000,000 or more cycles have no effect on their quiet operation, smooth and trouble-free service.

Constructed from the finest materials available, Dormeyer Solenoids are engineered to specifications to give maximum service in the most difficult industrial service.

Dormeyer Precision-Wound Coils— *Quality controlled and available in quantity through our modern mass production facilities. For every electrical purpose, including fine wire coils up to No. 50 Wire.*

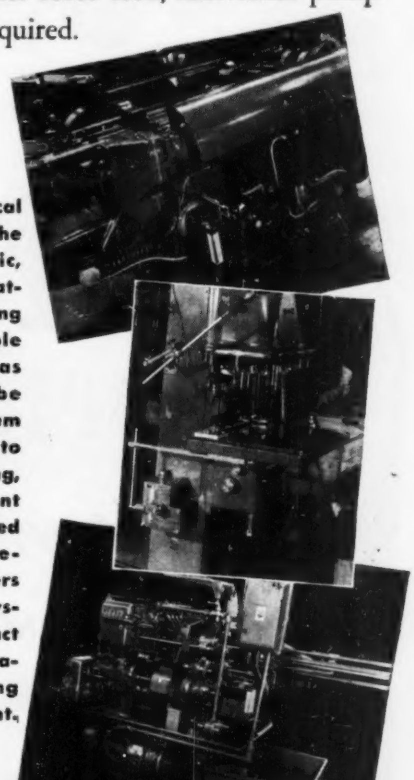
Send for catalog and include specifications for free estimate

DORMEYER INDUSTRIES
FACTORIES IN CHICAGO AND KENTLAND, INDIANA
3418 MILWAUKEE AVENUE • CHICAGO 41, ILLINOIS

Now PULSOLATOR

is made and sold by *Manzel*

Manzel has added the world-famed *Pulsolator* to its established line of Force Feed Lubricators and Chemical Feeders. Thus, this 55 year old organization now offers low pressure, single pump lubrication for applications where the unlimited pressure of Manzel force feed, individual pump systems is not required.

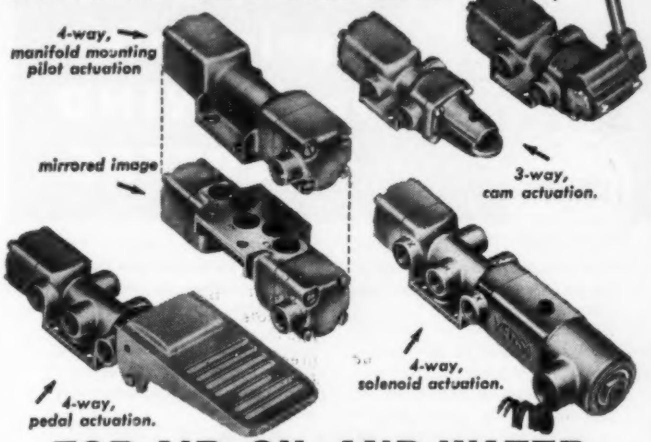


Here are three typical applications showing the *PULSOLATOR* automatic, loop systems for lubricating machine tools, printing presses and comparable equipment. As many as 1,000 bearings can be lubricated by one system from a loop line of up to 300 ft. Capacities, timing, and quantity of lubricant delivered can be adapted to individual requirements. Manzel engineers will be glad to plan a system to meet your exact needs either on new machines you are planning or on existing equipment.

Manzel

DIVISION OF FRONTIER INDUSTRIES
276 BABCOCK STREET, BUFFALO 10, NEW YORK

CONTROL VALVES IN ANY ACTUATION



FOR AIR, OIL AND WATER

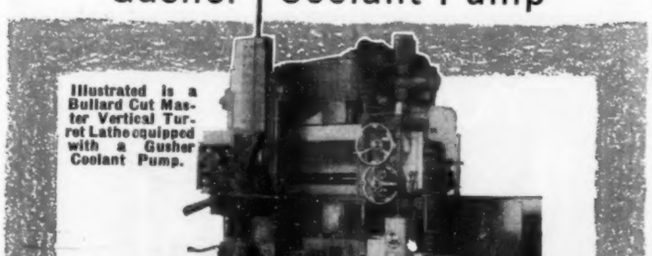
From over 1,000 standard combinations there is a type and size VERSA VALVE to suit every compressed air and low pressure water or oil hydraulic need, (0 to 200 psig). Interchangeable hand, foot, cam, pilot and solenoid actuating devices for 2-, 3-, and 4-way types in all popular sizes. Spring return, spring center or no spring action optional for any type actuation. All VERSA VALVES available in manifold mounting design.

Functionally designed to do a job, VERSA VALVES are unequalled for modern appearance and compactness. And any VERSA VALVE can be completely disassembled with an ordinary screw driver without disturbing pressure line connections.

SEND FOR YOUR FREE CATALOG AND DATA SHEETS

VERSA PRODUCTS COMPANY, INCORPORATED
249 SCHOLES STREET BROOKLYN, NEW YORK

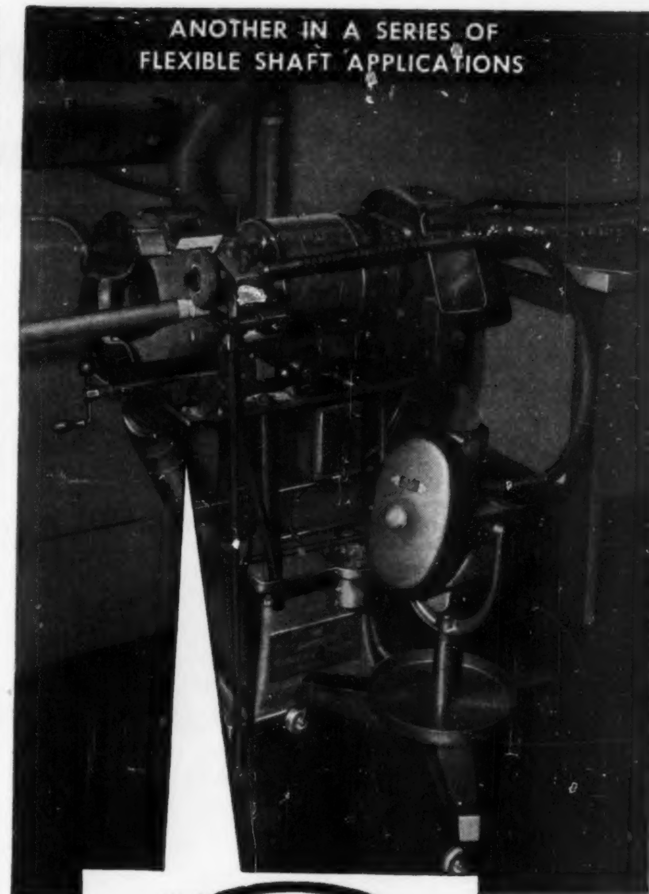
Low Cost Operation
is yours with a RUTHMAN Gusher Coolant Pump



Once a Ruthman Gusher Coolant Pump is installed it requires little or no attention. The pre-lubricated heavy-duty ball-bearings require no further lubrication, there are no packing glands to adjust. The dynamically balanced rotating shaft minimizes vibration and wear. Minimum power is consumed when the Gusher is throttled down. You are sure of low cost operation when you specify Gusher Pumps on your machine.

THE RUTHMAN MACHINERY CO.
1811 Reading Road Cincinnati, Ohio

ANOTHER IN A SERIES OF FLEXIBLE SHAFT APPLICATIONS



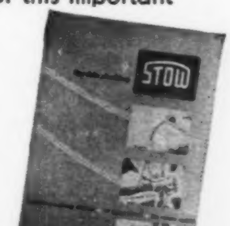
AT STOW WE PRACTICE WHAT WE PREACH

and save $$$ in production!

Concrete vibrator heads are subject to terrific tension load. Because of this, special precision grinding operations are necessary in their manufacture to insure that outside and inside diameter of casings are exactly concentric. By using a Stow Flexible Grinder in conjunction with another grinder, as shown in the illustration, Stow Manufacturing Co. saved 50% of the production time formerly required for this important operation.

send for free copy

Stow can save money for you, too. Write today for catalog and specific information.

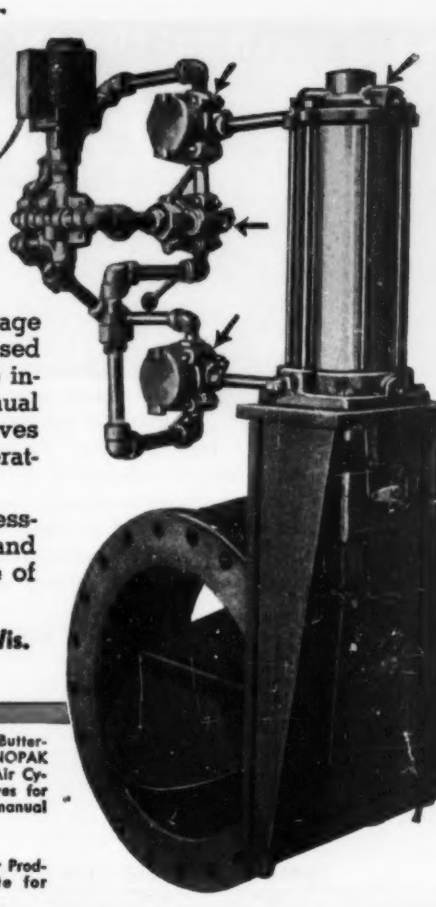


STOW MANUFACTURING CO.
11 SHEAR ST., BINGHAMTON, N.Y.

R-S Butterfly Valves

to Regulate Flow of Liquids and Gases,

. . . employ NOPAK *Cylinders controlled by* NOPAK *Operating Valves*



In the chemical, paper, petroleum and steel industries, in sewage plants, water works and other utilities, R-S Butterfly Valves are used for fast, accurate flow-control of liquids, vapors and gases. In the interest of speed and efficiency, and to reduce or eliminate manual effort, the R-S Products Corp. of Philadelphia equips many of its valves with NOPAK Air or Hydraulic Cylinders controlled by NOPAK Operating Valves.

The R-S No. 789 Butterfly Valve pictured here is typical. This successful application may suggest how you can use NOPAK Valves and Cylinders in your plant operations, or to improve the performance of equipment that you build for others.

GALLAND-HENNING NOPAK DIVISION • 2752 S. 31st St. • Milwaukee 46, Wis.

Representatives in Principal Cities

NOPAK
VALVES AND CYLINDERS
DESIGNED for AIR and HYDRAULIC SERVICE

R-S No. 789 Heavy Duty Butterfly Valve equipped with NOPAK Model "D" 8" x 12" Air Cylinder and NOPAK Valves for optional solenoid or manual valve control.

Refer to Sweet's File for Product Designers or write for Bulletin SW-2.

A 7555-½H

Engineered Air Regulation

☆

WHAT DO YOU REQUIRE OF AN AIR REGULATOR?

IS IT CAPACITY?

Kendall Model 10 Air Pressure Regulator gives up to 35 CFM with 90 PSI supply pressure

IS IT ACCURACY?

Increasing flow overcomes downstream line losses by jet tube venturi action in the Kendall Model 10

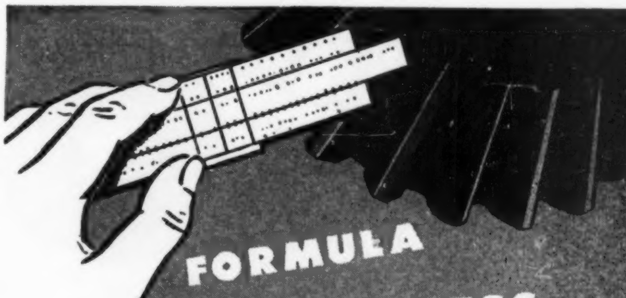
IS IT FREEDOM FROM SUPPLY PRESSURE INFLUENCE?

50 PSI variation in upstream pressure will result in less than 0.05 PSI change in downstream pressure on dead end service when you use a

KENDALL MODEL 10 AIR PRESSURE REGULATOR

AVAILABLE ¼" or ⅜" N. P. T.
RANGES 0-2, 10, 30, 60, 150 PSIG
Send for BULLETIN 4521

KENDALL CONTROLS CORPORATION
144 MOODY STREET WALTHAM 54, MASS.



FORMULA FOR SUCCESS

One proven way to build business is to give customers "extra" service at no extra cost. In ours, the gear business, for example, meeting the customer's requirements entails more than just filling his specifications. We're not satisfied until we've met all our own rigid standards of quality and dependability—no matter how much time and work is involved.

This added effort for each and every customer, we find, helps us keep our old customers and continually bring new ones into the fold. The Earle Gear & Machine Company, 4707 Stenton Ave., Philadelphia 44, Pa.

EARLE GEARS

It's good business to do business with EARLE!

been reading any advertising lately?

Chances are that you've been reading a lot of it in the last few minutes.

Did you get some ideas which might help solve some of your current design problems? This is a most common happening—nearly all readers of MD have the same experience. With another 20 seconds of your time, you can do something about it.

We'll be glad to help you.

Go back to that yellow page in the front of this issue and fill out one of those cards. We'll pay the postage, send your request on to the manufacturer. Without further ado, he'll be glad to send you the necessary information.

USE THE YELLOW CARDS ON PAGE 39

ENGINEERS

ELECTRICAL - MECHANICAL

Exceptionally fine opportunities exist in new medium-size research laboratory near Pittsburgh for development of small electro-mechanical apparatus.

Very attractive salaries will be paid for interesting professional work in any of these permanent positions:

Project Leader

to coordinate and supervise various phases of design, fabrication and testing of models in the development of precision apparatus for control of heating, cooking and cooling. Must have extensive experience in any or all of these fields.

Research and Development Engineers

for design, development and testing of controls used on heating and cooling equipment. Must be graduate E. E. or M. E. from recognized college.

High salaries will be commensurate with experience and ability. Very attractive employee benefits. Congenial working conditions. Good residential area near Research Center.

Please send resume of education, experience and salary history to:

P. O. BOX 489, IRWIN, PA.

ENGINEERS

AVAILABLE OR WANTED

WANTED: Engineers and Draftsmen. Progressive, well-known manufacturer of machine tools needs engineers and draftsmen for expanding machine and equipment design program. Company offers pleasant working conditions, stable employment and advancement opportunities. Location is in a pleasant Vermont town with all modern facilities. Send resume of personal and educational background and experience to Philip C. Durland, Bryant Chucking Grinder Company, Springfield, Vermont.

• • •

WANTED: Design and Process Development Engineers. Design and process development engineers wanted to work on developing and improving glass manufacturing processes for a leading glass company located in Ohio. Openings for mechanical, chemical, ceramic, and electrical engineers. Duties will involve carrying new processes through development into manufacturing. Salary commensurate with training and experience. Reply should include resume of training and education and should state salary requirements. Address Box 829 MACHINE DESIGN, Penton Bldg., Cleveland 13, O.

• • •

WANTED: Design Engineers for design of special medium weight automatic equipment. Will be called upon to experiment with and recommend improvements to existing equipment as well as design of new equipment. Degree in mechanical engineering desirable. Should have a minimum of five years of design experience. Location: Chicago. Compensation dependent upon experience and ability. Position is not dependent on defense contracts and is permanent and offers good working conditions with excellent company benefits. Replies will be treated confidential and arrangements will be made to interview selected applicants. Submit complete resume stating experience, education, salary required and date available. Address Box 828, MACHINE DESIGN, Penton Bldg., Cleveland 13, Ohio.

• • •

WANTED: Engineers-Machine Designers. We invite you to call or write for an interview. Inquiries held in confidence. Monarch Machine Tool Company, "Builders of Monarch Lathes", Sidney, O.

ADVERTISING INDEX

For Further Information On Anything Advertised In This Issue, Use The Yellow Cards—Page 39

ADVERTISING INDEX

For Further Information On Anything Advertised In This Issue, Use The Yellow Cards—Page 39